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A DESIGN PROCEDURE FOR MINIMIZING PROPELLER-INDUCED VIBRATION I--ETC(U)
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A DESIGN PROCEDURE FOR MINIMIZING PROPELLER-INDUCED VIBRATION IN HULL STRUCTURAL ELEMENTS

SSC-291 PROPELLER-INDUCED VIBRATION SEPTEMBER 1979

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SHIP STRUCTURE COMMITTEE

1979

Member Agencies:
United States Coast Guard
Naval Sea Systems Command
Military Sealift Command
Maritime Administration
United States Geological Survey
American Bureau of Shipping

Ship Structure Committee

An Interagency Advisory Committee
Dedicated to Improving the Structure of Ships

Address Correspondence to:
Secretary, Ship Structure Committee
U.S. Coast Guard Headquarters,(G-M/82)
Washington, D.C. 20590

SR-1240
September 1979

The rapid advance in ship size and power and the trend toward lighter hull scantlings prompted the Ship Structure Committee to investigate the propeller-induced vibrations in the hull and superstructure of the ship. High vibratory forces in the ship can cause discomfort in the living quarters, excessive "panting" type deflection of tank bulkheads, and fatigue cracks in webs and plating.

The first phase developed a bibliography published as SSC-281. It was made available for the October 15 - 16, 1978, Ship Vibration Symposium, sponsored jointly by the Ship Structure Committee and the Society of Naval Architects and Marine Engineers.

The subsequent phases, including the development of a design procedure for minimizing propeller-induced vibration in hull structural elements, have been completed and are reported here.

Henry H. Bell
Henry H. Bell
Rear Admiral, U.S. Coast Guard
Chairman, Ship Structure Committee

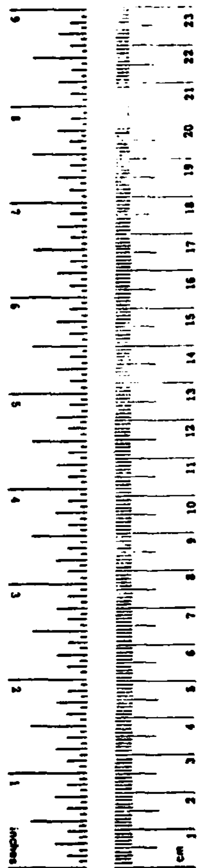
A

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

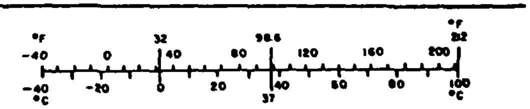
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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16. Abstract

A design procedure for minimizing propeller-induced vibration in hull structural elements is recommended. This procedure begins when the ship's vibration specifications are defined and continues through the design and construction process until the vibration levels measured during sea trials are compared with the specifications. Consideration is given to the hydrodynamic excitation and structural response of the propeller-induced vibration problem, with both analytical and experimental techniques being used in the design process. The recommended procedure is presented and discussed in the form of a flow diagram with 27 separate design steps. The process also contains five evaluation milestones. At these points, the design is assessed, and, if deficiencies are found, corrective action can be taken before the design proceeds. The recommended complete procedure is presented in this report for the first time. Many of the aspects of this procedure are still being developed, in particular, the influence of propeller cavitation on hull pressures and a simple but accurate treatment of water inertia. These indefinite aspects have to be treated empirically using judgment and experimental data. The portions of the procedure which are available are illustrated in an example using a single-screw, containerized and unitized cargo ship.

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TABLE OF NOMENCLATURE

A	cross-sectional area
A_m, A_n	areas associated with grid points m and n
a_m	acceleration of piston m
a_x	longitudinal clearance of propeller-hull, forward
a_z	vertical clearance of propeller-hull
C	factor reflecting density of fluid and proportions of pistons
c	speed of sound in fluid
c_c	critical damping coefficient
D	propeller diameter
$\frac{d(\ )}{dt}$	derivative of ( ) with respect to time
E	modulus of elasticity
F	force on sphere in direction of vibration
F_{mn}	mutual force between areas A_m and A_n
F_x, F_y, F_z	components of force in x, y, and z directions
f	frequency
g	hysteretic damping coefficient
I_y	moment of inertia about transverse axis
I_{yz}	product of inertia relative to horizontal and vertical axes
I_z	moment of inertia about vertical axis
J_x	torsional area constant about longitudinal axis
K	thrust bearing foundation stiffness
K_{P_1}	pressure coefficient for first harmonic
K_{P_2}	pressure coefficient for second harmonic
$K_{P_{1,2}}$	dimensionless pressure variable

K_T	thrust coefficient
K_{TB}	thrust block stiffness
$K_{xy}A$	shear area constant transverse plane
$K_{xz}A$	shear area constant vertical plane
L	length of ship
M_x, M_y, M_z	components of moment in x, y, and z directions
n	number of propeller revolutions per second
P, P_r, p	hull pressure
P_o	static pressure at centerline of propeller shaft at propeller; single amplitude of vertical component of propeller-exciting force in pounds (at blade frequency)
P_v	vapor pressure
R	radius
r	distance between dipole's center and location of desired pressure; radius from source
r_{mn}	minimum surface distance between grid points m and n on ship's hull
T	propeller thrust
t	time
V	velocity
V_a	axial velocity of water relative to propeller disc
V_m	velocity of model
$\cancel{V}$	volume
W	Taylor wake number
W_{max}	maximum Taylor wake number
Y	single amplitude in mils
$\bar{y}$	transverse coordinate of neutral axis
y', z'	coordinates of shear center of section

z	number of propeller blades
$\bar{z}$	vertical coordinate of neutral axis
α	angle of attack
Δ	displacement of ship in long tons
$\Delta P_1, (\Delta P_2)$	amplitude of first (second) component of averaged pressure fluctuation
$\Delta\alpha$	change in angle of attack
θ	angle between a vector to measuring point and force vector
ν	frequency in cycles per second
ρ	mass density of fluid
σ_n	cavitation index
$\phi_1, (\phi_2)$	phase angle of first (second) harmonic
ω	angular frequency
ωt	angular blade position ($\omega t = 0$ for blade in vertical, top position)
∇	displaced volume, m^3

I. INTRODUCTION

1. Overview of Program

At the outset of this study, it was recognized that various aspects of the propeller-induced, ship hull vibration problem had received attention in many previous investigations. These ranged in character from research-oriented technical papers (both theoretical and experimental), to some papers and reports that dealt with certain parts of ship hull design. Thus, the primary objective of this program has been to conduct a comprehensive survey and evaluation of these existing information sources and methods for predicting the exciting forces and the response of the hull structure to vibratory loads caused by the propeller system, and to recommend for design those procedures which appear both practical and reliable. In meeting this objective, the previously fragmented information is brought together into an overall design procedure which addresses the complete design problem.

This program has been carried out under four tasks. Initially a literature search was conducted to establish the state-of-the-art for prediction methods currently available. The results of this task served as the informational basis for this report, and have been previously published in the form of the bibliography identified in Reference 1. Emphasis in that document has been placed on the current generation of large, high-powered vessels, so that, of the approximately 550 bibliographic entries, over 60 percent were published since 1970. The second task dealt with evaluation of the design procedures. This has included principally a judgment about the suitability of the data for design purposes. The results of this task formed the basis for the third task, which required the recommendation of an overall design procedure and associated detailed parts. Application of this procedure to a specific ship was demonstrated in the final task. In essence, the results of all but the literature search are documented in this final report. However, emphasis is placed on the recommended procedure and additional practices appropriate for overall design, with very little explanation of why some previously used methods may have been omitted.

2. Definition of Propeller-Induced Hull Vibration Design Problem

In view of the fact that ship hull vibrations can be excited by a variety of energy sources, it is appropriate that some definition of the propeller-induced vibration problem be established, along with what general concepts will be included in the design procedures established for its solution. For simplicity, the problem to be addressed is shown conceptually in Figure 1. Thus, only those vibration responses excited by the propeller and its associated shafting are to be considered. Furthermore, a conceptual diagram of a desired design procedure is identified in Figure 2. Hence, the design procedure is to start with a given set of specifications, and progress with both analyses and tests to where the design has been validated by suitable model and full-scale sea trials.

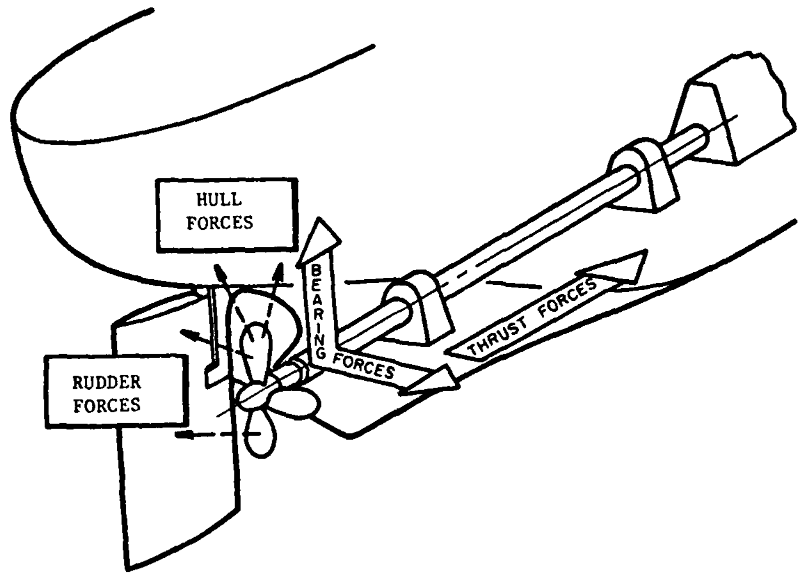


FIGURE 1. CONCEPTUAL IDENTIFICATION OF HUL.L VIBRATION SOURCES

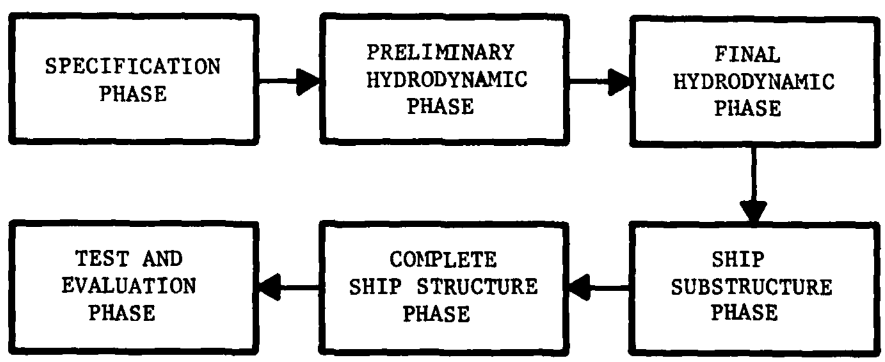


FIGURE 2. CONCEPTUAL DIAGRAM OF DESIRED DESIGN PROCEDURE

3. Consideration of Interdisciplinary Requirements

In view of the previously stated objective, it is obvious that development of a sufficiently general design procedure is a formidable task. This is especially true if it is to be applicable to many classes of ships. To be successful, the design process involves several different specialized naval architecture and marine engineering disciplines, as well as some others from traditional branches of engineering. Some areas included are:

- Naval Architecture and Marine Engineering
  - Ship Form Design
  - Propeller Design
  - Propulsion System Design
  - Rudder Design
- Theoretical Hydrodynamics
  - Propeller and Hull Loading
  - Cavitation
  - Boundary Layer and Potential Flow Theory (wake survey interpretation)
- Experimental Hydrodynamics
  - Wake Survey
  - Model Tests
  - Cavitation Tests
  - Hull Pressure Tests
- Structural Analysis
  - Propeller Shafts
  - Substructures
  - Main Hull and Superstructure
- Acoustics
  - Human Response
  - Equipment Response
- Experimental Vibration Force and Stress Measurements
  - With Shaker
  - In Service

It is obvious that no one person, and few engineering organizations, have complete expertise in all the above disciplines. However, a proper vibration analysis requires an understanding of the interrelationships between all of these factors and their coordination with the ship design procedure. Therefore, the design of a ship having acceptable vibration levels has been and will be established from the technical input of several sources. This is an important point. One should realize that a realistic recommended vibration design procedure must mesh with other ship design and construction processes. It is also important to consider the procedures in toto, and not merely in terms of one of its parts. For example, hydrodynamicists should not view the problem only in terms of a hydrodynamic solution; structural engineers should not view the problem only in terms of detuning the response from the excitation. Each group needs to realize the other's potential contribution to a solution and the necessity of incorporating input from all necessary sources.

II. DESCRIPTION OF RECOMMENDED OVERALL DESIGN PROCEDURE

In order to establish a recommended general design procedure for minimizing propeller-induced vibrations, it was necessary to adopt a philosophy on which the procedure would be based. Hence, a five-part design philosophy was formulated, as follows:

(1) Vibration Specifications Should be Quantitatively Defined with Attention Given to Human Exposure, Machinery and Equipment, and Structural Strength.

(2) Excitation Forces Should be Kept to a Minimum.

(3) Structural Resonances with Propeller Excitations Should be Avoided.

(4) Vibration Response Levels Should be Measured During Sea Trials to Ensure Specifications are not Exceeded.

(5) Measured Vibration Levels Should be Compared with Predicted Values to Assess Design Procedures.

The significance of this design philosophy will become more apparent when details of the design procedure are discussed. However, a few general comments are in order at this time. It is clear that for a ship hull design to be successful, there must first be selected a reasonable set of design criteria, or goals, on which the process is to be based. This is the purpose of Item 1 of the design philosophy. These specifications should be established in the ship's design contract and serve as a standard in guiding the design process. Vibrations levels recorded during sea trials can also be compared with the specifications to judge the ship's acceptability from a vibration point of view.

It is almost axiomatic that excitation forces should be kept to a minimum, as stated in Item 2. A propeller mounted far aft of a ship's stern may induce very little ship vibration, but this solution is not very practical in terms of propeller efficiency, propeller whirl, and other associated structural problems. What is meant is that attention should be given to those factors which can reduce the excitation, e.g., stern configuration, propeller geometry and clearances, propeller wake, and cavitation.

There are many components of a ship's structure which can be excited by the propeller-generated forces and pressures. They include vibrations associated with the lateral, longitudinal, and torsional response of the propulsion system; overall vertical bending and coupled lateral-torsional bending of the ship hull; vibrations of major substructures such as the engine room, machinery spaces, and superstructures; and response of local structures such as the rudder and local plating. To make matters more complicated, each of the above systems is coupled to some degree to the others. One of the primary objectives of the design procedure outlined in

this report is to be able to predict accurately the various structural resonances of the ship and determine if they will be excited by the propeller. If so, these resonances should be avoided because they will result in large amplifications. This is the reasoning behind the design philosophy presented in Item 3.

Item 4 is normally part of the sea trials for the acceptance of the vessel by the ship owner. The only additional comment which should be made in this report is that vibration levels should be measured at the critical locations throughout the ship. For human exposure, these include living quarters, watch stations, steering gear spaces, machinery spaces, and cargo spaces. For machinery and equipment, the longitudinal, lateral, and torsional vibration levels of the propulsion system should be measured along with those of any other critical components. Finally, the vibrational stresses in critical structural locations should be monitored to ensure that fatigue endurance limits are not exceeded. Critical areas would include, for example, bottom framing over the propeller, rudder and rudder horn, stern bearing support, vertical columns on intersecting bulkheads, and masts and spars.

The purpose of Item 5, in which the measured vibration levels are compared with the predicted values, is to assess the validity of the design procedure. It is extremely important to conduct this post-mortem analysis because it allows the entire design process to be critically reviewed to determine its strong and weak points. For example, if unacceptable vibrations were measured on the bridge where none were predicted by the analysis, the fault probably lies with inadequate structural modeling techniques of the superstructure. If the stern plating vibrates at the correct frequency, but at greater amplitudes than predicted, the problem could be traced to underestimation of the propeller-generated pressures by the hydrodynamic computer code or the influence of cavitation.

Having the previously defined philosophy in mind, we now introduce in Figure 3 a flow diagram of the recommended design procedures for minimizing propeller-induced vibrations. The procedure consists of twenty-seven individual blocks ranging in time from the establishment of vibration specifications to after the sea trials are conducted. Each one of these individual blocks will be discussed in detail in Chapter III. In discussing these sub-procedures, it is the intent not only to give perspective to the function and purpose of each block, but to present detailed information on how each can be used in the design process. This can best be accomplished using tables summarizing the pertinent information.

The overall procedure is divided into six design phases: (1) specification, (2) preliminary hydrodynamic, (3) final hydrodynamic, (4) ship substructure, (5) complete ship structure, and (6) test and evaluation. The purpose of these phase designations is simply to give a qualitative description of the overall design process in accordance with the design which was given in the Introduction. Figure 3 also shows five evaluation milestones which are located approximately at the end of each of the last five design phases. The purpose of these evaluation milestones is to provide a means of assessing the design integrity up to that point. If it is acceptable, the design may continue on to the next phase; if not,

DESIGN

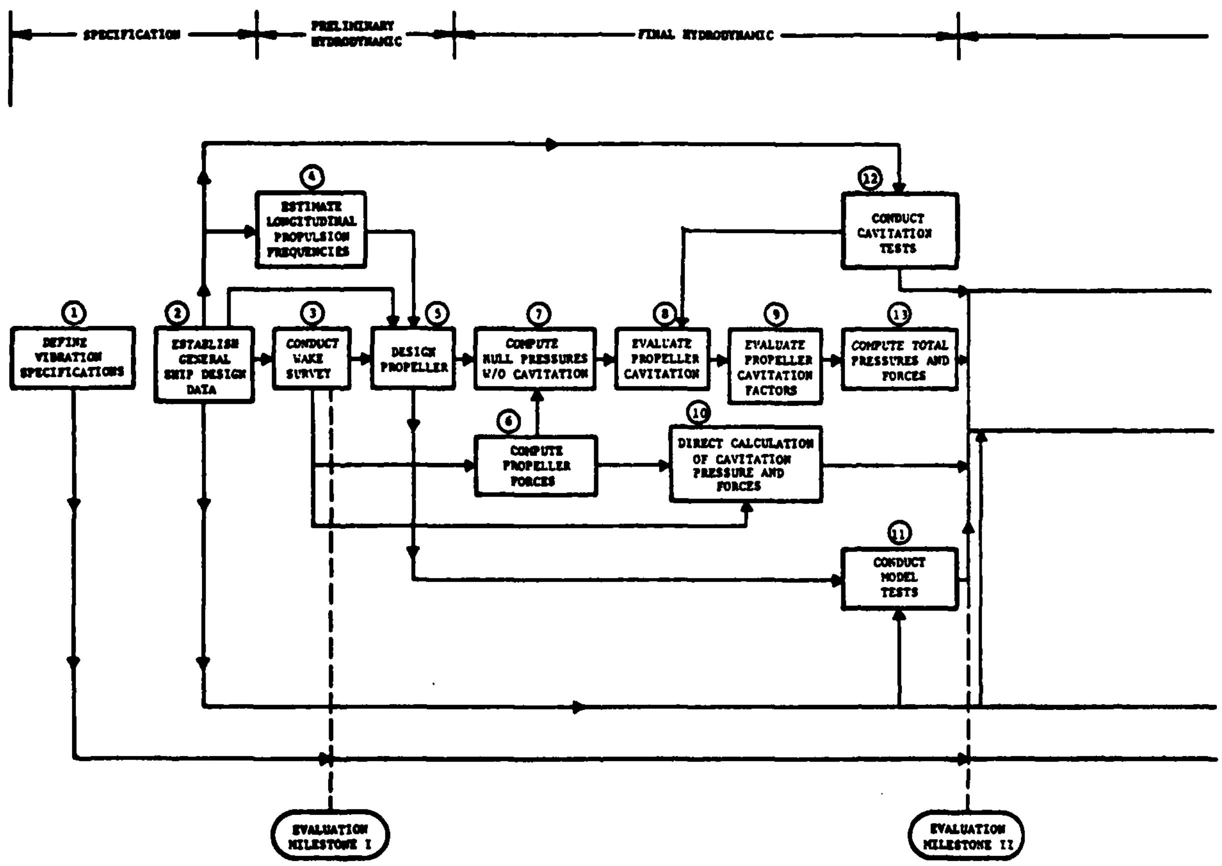
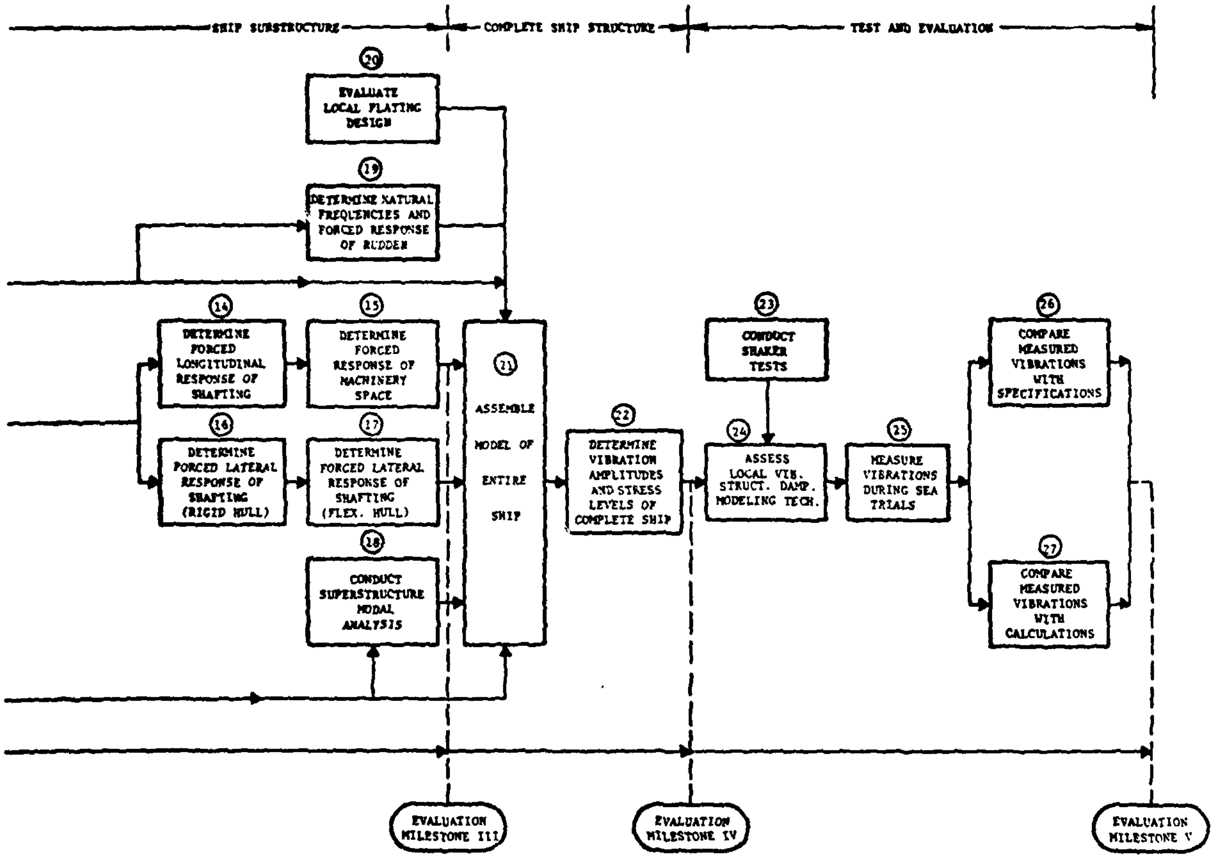


FIGURE 3. FLOW DIAGRAM OF RECOMMENDED DESIGN

PHASES



PROCEDURES TO MINIMIZE PROPELLER-INDUCED VIBRATIONS

corrective action should be taken before the process continues. The advantage in having these frequent evaluations is that potential problem areas can be identified and corrected early in the overall process. This, hopefully, will avoid the all-too-frequent problem of having a certain design fixed, with changes possible only through costly modifications. The evaluation milestones will be discussed in Chapter IV.

III. DETAILED STEPS FOR SHIP VIBRATION DESIGN

The procedures associated with individual design steps identified as blocks in Figure 3 will now be discussed in detail.

1. Define Vibration Specifications

It is the goal of the entire vibration design procedure to secure a ship which has a minimum of vibration. This goal cannot be reached, however, without clearly establishing what vibration levels are and are not acceptable to the shipowner. These levels must not be arbitrarily set, but must be within certain limits to ensure safe and efficient operation of the ship. Too stringent conditions impose an unwarranted burden on the shipbuilder and high design and construction cost, while the ship may vibrate badly if the specified levels are too high.

Undesirable levels of shipboard vibration manifest themselves in one or more of the following areas:

- Human Exposure and Habitability
- Machinery and Equipment
- Structural Strength

Of these three, experience with the current generation of ships clearly shows that the greatest problem lies with human exposure and habitability. This is due to the increased size, horsepower and speed of the vessel, and the greater attention paid to the health and comfort of the crew. Reference 2 discusses in some detail the problem of shipboard vibration and its effect on habitability. It is important to note that acceptable levels of vibration for human exposure can be different in different portions of the ship. For example, continuous exposure levels must be maintained in the living quarters and watch stations, while less stringent requirements would be imposed in areas such as the steering gear, cargo, and machinery spaces.

The machinery which is affected by propeller-induced vibrations is usually associated with the ship propulsion system. Included are the longitudinal, lateral, and torsional vibrations of the shafting system and vibration in the main power plant. Other items of equipment particularly sensitive to vibration would include those associated with navigation, communication, or special cargo which the ship is carrying. However, all machinery and equipment should be able to withstand levels which are acceptable to humans.

The final way in which vibrations can be detrimental to the performance of a ship is by reducing its structural strength through fatigue. This is particularly a problem at highly loaded areas in the ship which experience many stress reversals. Such areas would include the bottom framing over the propeller, the rudder and rudder horn, the stern bearing support, vertical columns on intersecting bulkheads, and masts and spars.

If we turn now to the set of specifications, they should include as a minimum the following items:

1. Limits of acceptable vibration for human exposure, machinery and equipment, and structural strength.

2. The types of experimental and analytical studies which must be performed to ensure the requirements are met and the extent of the documentation for these studies.

3. The test requirements and methods for vibration measurements during the acceptance trials.

4. The responsibility for correcting vibration problems should they occur during the acceptance trials or during the subsequent warranty period of the ship.

From the literature it appears that the limits of acceptable vibration in humans are well established. Reed [3], in a 1973 paper, discussed the question of acceptable vibration levels and pointed out that the International Standard ISO 2631, "Guide for the Evaluation of Human Exposure to Whole-Body Vibration" [4], published by the International Organization for Standardization, provides an excellent base for setting these levels on ships. The standard permits vibration levels to be rated numerically as percentages of the established standard of fatigue-decreased proficiency. The standard is related to frequency, direction of motion, and the exposure time at the different locations in the ship. Safe exposure limits and reduced comfort limits are defined in terms of percentages of the fatigue-decreased proficiency level. This ISO Standard also has been adopted by the American National Standards Institute, and it appears the Standard can be used to establish rational vibration limits for human exposure.

Guidance for acceptable vibration of marine steam and heavy-duty gas-turbine main and auxiliary machinery plants has been published recently by The Society of Naval Architects and Marine Engineers (SNAME) [5]. It was prepared by Panel M-20 (Machinery Vibrations) of the Ships' Machinery Committee and was intended to serve as a reference standard in ship's specifications and procurement documents for new marine equipment. This Code C-5 presents in detail the vibration limits of the machinery plants as well as those for the longitudinal, lateral, and torsional response of the propulsion system. It also specifies what type of tests are to be conducted and the instrumentation required to measure the vibration levels.

SNAME also has two additional codes dealing with vibration measurements. The first, Code C-1 [6], is concerned with (1) vibration of the ship girder excited by the propulsion system at shaft frequency, harmonics of the propeller-blade frequency, and frequencies associated with major components of machinery; and (2) vibration caused by propeller excitation of the propulsion shaft system. The second, Code C-4 [7], addresses local vibration of ship structural elements such as the deckhouse,

decks, bulkheads, masts, machines, foundations, or other appurtenant elements of interest. Both of these Codes can be referenced in the ship specifications as to the manner in which vibration measurements will be made.

The final item in the specification concerns structural strength. Since the ship's structure is least affected by propeller-induced vibration, little attention has been received in this area. The specification should state that the stresses in structural locations subjected to high loadings should not exceed the fatigue stress endurance limit of the material with an appropriate factor of safety. Typical critical areas were mentioned earlier in this section; these include the bottom framing over the propeller, the stern bearing support, and masts and spars. Stresses in these locations could be measured with strain gages during the acceptance trials.

2. Establish General Ship Design Data

In any design process there must be a starting point at which basic information is assumed to be known. For the complete ship design, nothing more specific than the mission of the vessel would be given. This is too early to seriously consider the problems associated with propeller-induced vibrations. More information about the ship's size, configuration, and operating requirements must first be established in the feasibility studies.

The procedures presented and discussed in this report assume that certain general ship design data are available. The amount of information required is the minimum necessary to begin the design procedure. As additional ship data become known from other segments of the total design process, they will be used in the vibration study. One should also realize that this information is preliminary and may be altered if the design procedures show changes are necessary.

This study will assume that the preliminary design data necessary for approval of the basic design by the Maritime Administration are available. Such data would include:

- Preliminary Lines Plan
- Preliminary Midships Section Based on ABS Rules
- Preliminary General Arrangements of Decks and Inboard and Outboard Profiles
- Preliminary Weight and Center of Gravity Estimates
- Speed and Power Estimates (No Model Tests)
- Preliminary Machinery Arrangements
- Preliminary Capacity Plan
- Preliminary Hydrostatic Curves

- Preliminary Flooding Curves and Damaged Stability Calculations

- Preliminary Specifications Indicating Propeller RPM and Diameter

This report will discuss those recommended design procedures which can be used after the preliminary design has been completed. This is not to imply that the preliminary design should not consider the problem of propeller-induced vibrations. It is simply more difficult to quantitatively establish the vibration characteristics of a given ship because all the necessary elements are not yet defined. Instead, the preliminary design must rely heavily on the experience of the naval architects and existing rules from the classification societies. Insight as to whether a ship will develop vibration problems can often be inferred by the excitation and response levels on ships having similar stern lines, propeller RPM and power, machinery arrangement, and general structural configuration.

Table 1 shows the data which are necessary to begin the design procedure presented in Figure 3. As the entire ship's design progresses, these data will be supplemented by additional information when it becomes known.

TABLE 1. SUMMARY OF DESIGN BLOCK 2--
ESTABLISH GENERAL SHIP DESIGN DATA

Input Data	To Develop	Required for
Ship Power and Speed	Hydrodynamic Test Model Definition	Wake Survey
	Estimate of Propeller and Shaft Sizes	Propeller Design and Shaft RPM
Preliminary Scantling and Shafting Plans	Evaluation of Thrust Bearing and Location of Bearings	Longitudinal and Lateral Analysis of Shafting
	Formulation of Structural Model Mass and Stiffness	Entire Ship Vibration Analysis
Ship Lines	Hydrodynamic Test Model Definition	Wake Survey
	Hydrodynamic Test Model Definition	Cavitation Tests Behind Ship Model

3. Conduct Wake Survey

After the general ship design data, including the ship's lines, have been established, the next task is the conduct of model tests. These tests are basically used to confirm data which were predicted in the preliminary hydrodynamic design. However, as seen from Figure 3, the model tests are continued in time until all hydrodynamic work is completed. The latter model tests are not only used to confirm predictions made in the final hydrodynamic design phase, but also to obtain data not available through current analysis techniques. This is generally in the area of propeller cavitation and cavitation-generated pressures.

Some of the data which can be obtained from a complete set of model tests include:

(a) Resistance or EHP versus speed, with and without appendages (usually done)

(b) Sinkage and trim of the hull (usually done)

(c) Wave profile and flow pattern around hull (sometimes done)

(d) Shaft horsepower and RPM versus speed for fully appended hull. Determined in early tests with stock propeller, then with propeller designed for the hull (usually done).

(e) Wavegoing performance of the hull (sometimes done)

(f) Shallow-water and restricted-channel behavior (rarely done)

(g) Dynamic stability, maneuvering characteristics, and controllability when backing (often done)

(h) Wake vectors without appendages to serve as a guide for proper appendage location and orientation (often done)

(i) Wake vectors in way of propeller disc with appendages located (often done)

(j) Open-water and cavitation data on propellers designed especially for hull (usually done)

(k) Nature and magnitude of the propeller vibratory forces imposed on hull (rarely done)

These items cover the entire area of model testing, and as indicated, not all tests are conducted for every ship. Items i, j, and k relate directly to the propeller-excited ship vibration problem, and each will be discussed in the appropriate design block.

As far as the wake survey is concerned, its primary purpose is to provide data necessary for the propeller design, the computation of propeller and hull forces, and an evaluation of the extent of propeller cavitation.

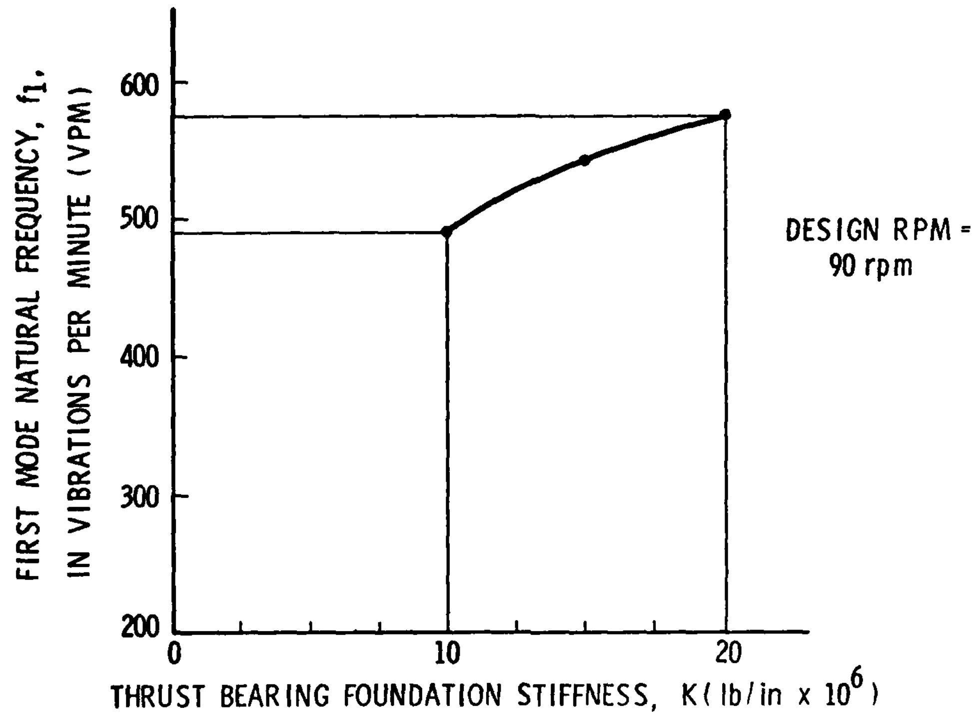
The wake, when determined in absence of the propeller, is called the nominal wake field. Van Oossanen [8] points out that it is becoming common practice to no longer accept the measured wake behind a model in a towing basin as representative of the full-scale wake field. Differences arise because this nominal wake does not consider the effects of the propeller on the true wake and because Reynolds number scaling is not included. In the last few years, there have been attempts to include these effects by numerical calculations. Hoekstra [9] at the Netherlands Ship Model Basin has developed a procedure to account for both Reynolds number scaling and the effect of the propeller on the nominal wake field.

Reference 10 presents the results of a recent British Ship Research Association project on propeller-excited vibrations in which methods of wake quality assessment are discussed. The authors point out quite correctly that the main cause of unsteady cavitation and large propeller bearing forces is the non-uniformity of flow into the propeller. It is therefore of the utmost importance that this wake be measured and evaluated accurately.

Until recently, only the axial component of the wake velocity was measured. This was partly because the available experimental techniques could only measure one component at a time and because existing hydrodynamic computational techniques did not include tangential and radial velocities. However, at the present time, all three components can be measured with a five-hole pitot tube. This has stimulated additional research into the ways the information concerning the three-dimensional velocity field can be used in analytical prediction techniques. Other experimental techniques which are used to obtain the wake are the hot-wire anemometer and the laser-Doppler anomometer. These are discussed briefly in a paper by van Gent and van Oossanen [11].

4. Estimate Longitudinal Propulsion Frequencies

In general, to keep propeller and hull excitation forces low, it is desirable to use many blades on the propeller. The number of blades chosen is set primarily by the natural frequency of the shafting and propeller in longitudinal vibration. To ascertain the probable frequency that will be found after the design of the propulsion system and its supports are developed, it is useful to have a plot of natural frequency versus foundation stiffness such as shown in Figure 4. Using values of the probable range of foundation and thrust bearing stiffness, the probable range of shaft longitudinal frequency is determined. The number of blades for the propeller is chosen so that, preferably, the excitation frequency is less than 80 percent of a possible propulsion natural frequency. A less desirable, but sometimes necessary, solution is to locate the longitudinal natural frequency f_1 about 30 percent below the blade frequency of the minimum steady operating speed. Figure 4 shows that, if the foundation stiffness K is estimated to be between 10×10^6 and 20×10^6 lb/in, then a four-bladed propeller will satisfy the criterion over the entire stiffness range. The five-bladed propeller's natural frequencies all are above the excitation frequency, but if the actual foundation stiffness turns out to be close to 10×10^6 lb/in, undesirable vibrations could develop. A six-bladed propeller would not be acceptable because the shaft would pass through resonance for the lower



K (lb/in)	f_1 (VPM)	MAIN SHAFT SPEED (rpm) 4 BLADES	5 BLADES	6 BLADES
10×10^6	494	124	99	82
15×10^6	542	136	108	90
20×10^6	571	143	114	95

FIGURE 4. FIRST MODE LONGITUDINAL NATURAL FREQUENCY VERSUS THRUST BEARING FOUNDATION STIFFNESS

foundation stiffness and would lie dangerously close to resonance for the remaining values.

Estimates for the thrust bearing foundation stiffness can be found in the recent SNAME Technical and Research Report R-15 [12] and in the earlier work by Kane and McGoldrick [13].

For making these predictions, the power and machinery arrangements of the plant must be defined. These, along with the propeller RPM or diameter, will have been specified by the preliminary design data. From this, the approximate propeller weight and water inertia associated with longitudinal vibration can be established by the techniques given by Mott [14] and Lewis and Auslaender [15]. Also, the approximate diameter of the tailshaft and lineshaft can be established by rules of the various ship classification societies.

The simplest procedure for predicting the natural frequencies is on the basis of a one-degree-of-freedom system consisting of the propeller and water inertia plus a portion of the shaft mass vibrating against the stiffness of the thrust bearing and its foundation. Since the shafting weighs considerably more than the propeller and adds flexibility, this procedure is not very good.

An improved procedure is to model the propeller and shaft as a series of concentrated masses and elastic elements and use a Holzer process for frequency computation. With this degree of complication, it becomes desirable to use one of the many digital computer programs available. These programs are usually based on finite-element or finite-difference methods, and several of the programs are discussed in Reference 16. With the high degree of sophistication and accuracy found in commercially available structural analysis programs, the choice of a particular code is governed by its convenience and cost to the user.

Table 2 presents a summary of the purpose, the input and output information, and pertinent references for this design block. Its format is typical of the tables for the remaining design steps and is intended to provide the reader with a concise summary of the individual procedure. The references listed in the tables are by no means exhaustive, but are particularly useful for design purposes. Reference 1 provides a more exhaustive list.

5. Design Propeller

After the number of blades has been selected based on the results of the longitudinal propulsion frequency analysis, the next step is to establish the propeller design. The primary purpose in this step is to select the propeller geometry which will provide the ship the highest propeller efficiency for the specified operating conditions. The design of the propeller must also consider ancillary problems such as blade strength and deformation, as well as selection of propeller materials and coatings to resist corrosion and erosion.

TABLE 2. SUMMARY OF DESIGN BLOCK 4--ESTIMATE LONGITUDINAL PROPULSION FREQUENCIES

Purpose:	To Establish the Number of Propeller Blades so that the Blade Rate Frequencies are Removed from Longitudinal Shafting Natural Frequencies
Input:	Propeller RPM Horsepower Machinery Arrangements Shafting Diameter Approximate Propeller Diameter Estimates of Propeller Weight and Water Inertia Range of Thrust Bearing and Foundation Stiffness
Output:	Recommended Number of Propeller Blades
References:	12 - 15

Propeller design is a highly specialized field, and selection is usually based on the recommendations of a consultant or a company active in propeller design. For these reasons, this report will not attempt to make other than general comments as to the propeller design process. Reference 17 provides excellent background information, while References 18-20 discuss a few of the current techniques used for propeller design.

The propeller design primarily influences the ship's vibration levels through the number of blades. It was for this reason that the longitudinal shafting frequency analysis was conducted in Block 4. Generally, there is little change in efficiency between, for example, a three-, four-, five-, and six-bladed propeller, and the final selection may be based upon vibration considerations. As a general rule, increasing the number of blades usually lowers the excitation forces on the shafting and the fluid pressures transmitted to the ship's hull. There is a tradeoff, however, because increasing the number of blades also increases the possible number of resonances with the hull and the propulsion system.

The amount of propeller skew also influences the vibration levels experienced in the ship. Generally, as the skew back of the propeller increases, the bearing forces as well as the surface pressures decrease. The axial vibratory forces and torques generated by the propeller decrease rapidly, and the vertical and lateral forces and moments generally, but not necessarily, decrease. Surface pressures also decrease, sometimes quite significantly. The decrease in surface forces comes from both the contribution of cavitating and noncavitating propeller pressures. Reference 21 presents theoretical and experimental data which show the advantages

and disadvantages of highly skewed propellers as compared with conventional propeller design. The design procedure and model evaluation techniques used by the Naval Ship Research and Development Center for a highly skewed propeller for a cargo ship are discussed in Reference 22.

Design of the propeller is really more a part of the ship design process than the ship vibration analysis, although there is a small input from the hull vibration process.

Table 3 shown below summarizes the data required for the propeller design process.

TABLE 3. SUMMARY OF DESIGN BLOCK 5--
DESIGN PROPELLER

Purpose:	To Design a Propeller for the Given Ship Which Will Produce the Highest Efficiency
Input:	Power Requirements Ship Speed Propeller RPM Wake Data Propeller Diameter Limitation Estimate of Propeller Skew Number of Blades
Output:	Geometric Form of Propeller
References:	17 - 22

6. Compute Propeller Forces

Prior to about 1960, the determination of propeller forces was by measurements on models, primarily by Frank M. Lewis [23]. In the late 1950's estimates began to be made on a quasi-steady-state basis using the procedures developed by Burrill [24] for evaluating the loading and efficiency of propellers whose circumferentially averaged wake varied along the propeller radii. A computer program for calculating the harmonic forces and moments generated by the propeller working in varying wakes based upon this quasi-steady-state procedure was applied by Hinterthon [25]. A similar computer program, also based on Burrill's procedure, but including as well the Theodorsen effects (i.e., the inertia of the fluid in responding to circulation changes resulting from changes in angle of attack), was developed by CONESCO [26]. The first tends to give high values of harmonic forces and moments and errors in their phase because the inertia effects are neglected. The latter program tends to give low values of harmonic forces and moments because the steady-state solution assumes flow over the

tip and interaction between blades that are not developed in the unsteady flow. Both of these programs have been superseded by improved analysis of the problem.

In 1958, Ritger and Breslin developed a theory for the unsteady thrust and torque of a propeller in a ship wake based upon unsteady airfoil theory. This work has been continued by Tsakonas and Jacobs [27] and is now a fully developed program for predicting the harmonic forces and moments exerted by a propeller on its supporting shaft, when working in the wake behind a ship. This program is based upon lifting surface theory. Although the computations are long, they are easily handled by a computer. A description of the program is given in Appendix A-1. This program is widely used both in the United States and abroad.

The Department of Naval Architecture and Marine Engineering at the Massachusetts Institute of Technology has also been active in the prediction of the harmonic forces and moments transmitted by a propeller to its supporting shaft. Using unsteady flow theory with the propeller blade represented as a lifting line, Neal A. Brown developed relations for determining the periodic propeller forces [28]. Several computer programs based on this theory have been developed. They are presented in Appendices A-2 and A-3.

More recently, Kerwin and Frydelund [29] have approached the unsteady force problem with another procedure. It is a discrete element approach for the computation of unsteady blade pressure distribution in the absence of cavitation. The work is still ongoing, and plans are to extend the procedure to include the effects of unsteady cavitation. A discussion of Kerwin's computer program is presented in Appendix A-4.

Similar procedures to those developed in the United States have been developed in Europe. M. T. Murray and J. E. Tubby [30] at the Admiralty Research Laboratory developed a computer program for determining the unsteady shaft forces from propellers. Information on this is presented in Appendix A-5.

Table 4 shows the basic information required to determine the hydrodynamic forces and moments acting on the propeller. Some of the input data may vary slightly, depending on the particular computer program used. For more detailed information, the reader is referred to the listed references. It should also be noted that all input information is available at this point in the design stage, either from the ship's operating condition, propeller design, or wake survey.

The results of the numerical computations provide the mean and harmonics of the blade frequency forces and moments, usually in the longitudinal, vertical, and lateral directions. These forces and moments can be applied to a structural model of the ship to determine its forced response to propeller excitation. This procedure will be discussed in the ship substructure and complete ship structure design phases.

It would be very advantageous, at this point in the ship's design, to be able to estimate the amplitudes of stern vibration based on the

TABLE 4. SUMMARY OF DESIGN BLOCK 6--
COMPUTE PROPELLER FORCES

Purpose:	To Predict the Hydrodynamic Forces and Moments Acting on the Propeller
Input:	Propeller Blade Geometry Propeller Geometry Ship Speed Results of Wake Survey (Usually Given in Terms of the Fourier Coefficients of the Spatial Variation of the Axial and Tangential Components of Wake)
Output:	Mean and Blade Frequency Force and Moment Components (This Information is Usually Given for Three Orthogonal Forces and Moments) The Time Varying Blade Pressure Distribution at Each Wake Harmonic
References:	27 - 32

computed propeller forces. This would allow the designer to predict if excessive vibration levels are probable and to take corrective action before the design proceeds any farther. Unfortunately, no such general estimation technique exists, but McGoldrick [31] does give empirical formulas for the vertical displacement and torsional rotation of the stern. The formulas are applicable only when the blade frequencies fall well above the range of significant hull mode frequencies. McGoldrick gives the formula for vertical vibration as

$$Y = \frac{P_o}{3.4 \times 10^{-6} \times \Delta \times (cpm)^2}$$

where

Y = the single amplitude in mils (a mil is equal to 0.001 in.)

P_o = the single amplitude of the vertical component of propeller-exciting force in pounds (at blade frequency)

Δ = the displacement of the ship in long tons

cpm = the blade frequency in cycles per minute

The empirical constant in the formulas is the factor 3.4×10^{-6} which was obtained by shaker tests conducted on the *SS Gopher Mariner*. McGoldrick indicates that there is some reason to expect the empirical constants

could be used for different ships, but that much more experimental data are needed to establish the constant for various classes of ships. To date, this has not been done. It should again be emphasized that procedures for estimating the stern vibration levels based only on the propeller forces and gross ship properties are needed at this point in the design phase. Development of such techniques is certainly worthy of further investigation.

7. Compute Hull Pressures Without Cavitation

It is the purpose of Design Blocks 7-10 to compute the excitation pressures on the ship's hull, including the effects of cavitation. The most direct method would involve calculating the type and extent of cavitation on the propeller operating in a given wake and then computing the fluid pressures generated on the hull. In the United States, no such direct procedure is available, although research is being conducted in this area. Van Oossanen reports [33] that procedures for calculating these hull cavitation pressures are available at the Netherlands Ship Model Basin and are used for design. They will be discussed in Block 11.

In this country, the procedure is roundabout and consists of predicting the hull forces generated by the propeller in the absence of cavitation and modifying the results by an empirical factor to account for cavitation effects. The factor is determined by (1) experience with full-scale measurements, (2) model tests in a cavitation towing tank, and (3) model tests in a cavitation tunnel of sufficient size to include modeling a portion of the ship.

In a sense, the use of a factor applied to the pressures determined in the noncavitating condition is theoretically unsound because the pressures are generated by another mechanism other than that responsible for the pressures generated in the noncavitating case. The hull pressures are generated as the sum of three different processes. The first is the pressure due to propeller loads, i.e., the difference in pressure on the face and back of the blades. The second source of pressure generation is the passage of the propeller blade bulk through the water. Generally the pressures from these two sources are approximately equal in amplitude, but can be quite different in phase. The third source, cavitation, is the growth and decay of cavitation bubbles as the blade moves into high wake regions. Since the growth and decay of a volume radiate pressure much more effectively than moving a volume from one place to another or introducing a flow from a source to a sink, the pressures from small cavitation volume changes can be large.

The determination of propeller-generated hull forces can be made by two processes: (1) estimation of the hull pressure and (2) an integration process involving Green's function which yields the total excitation force. Either process involves many engineering approximations for a reasonable solution. Generally the hull pressure process involves determining the pressure that would be generated by the loading and thickness of the propeller in a free field and multiplying this pressure by a factor to represent the pressure of the hull. This so-called boundary factor is 2 for an infinite flat plate located adjacent to a working propeller because the restraint on the flow caused by the plate can be simulated by an "image"

propeller working symmetrically opposite to the real propeller. The free-field pressure is that induced from both propellers, which is twice that of one. The value of 2 is normally used with reasonable accuracy for points on a ship's hull. References 34 and 35 indicate, however, that comparison between the calculated and measured hull pressures showed somewhat large variations from the value of 2 for cavitating and noncavitating propellers. This process is entirely inadequate for estimating differential pressures across narrow surfaces such as skegs or rudders. The Green's function process requires an estimate of the added mass of the hull surface for motions corresponding to each of the components of force and moment that are required [36]. Theoretical processes for predicting pressure differences across wedge and cone shaped surfaces [37,38] are available, but not yet programmed. A theoretical approach, the Smith-Hesse procedure, for predicting the hull pressure is available, but the calculation is so long that it has not at the present time been programmed.

In Reference 39 some of the methods available for calculating the pressure field around a propeller in a free stream are discussed and compared to cases where experimental results are available. The results are only valid for the noncavitating propeller, and Reference 39 indicates the results are changed appreciably when extensive cavitation is present.

Table 5 presents the data generally required for the computation of the hull forces or pressures. Some of the information will vary, depending on the particular computer program used. Procedures for predicting hull pressures by lifting line and lifting surface theory are presented in Appendices B-1 and B-2, respectively. Appendix B-3 presents the procedure

TABLE 5. SUMMARY OF DESIGN BLOCK 7--COMPUTE HULL PRESSURES WITHOUT CAVITATION

Purpose:	To Compute the Excitation Levels without Cavitation Acting on the Hull at the Ship's Stern
Input:	Propeller Blade Geometry Wake Distribution The Spatial Location of Points on the Stern where Pressures are Desired The Steady and Time-Dependent Blade Loading Distributions
Output:	Steady and Harmonic Components of the Pressure Field Generated by a Noncavitating Propeller. (For the Green's function method, all components of the total hull forces and moments at multiples of the propeller blade rates can be computed)
References:	27, 32, 36, 40, 41, 42

for computing the total excitation force on the ship's stern by using the Green's function approach.

8. Evaluate Propeller Cavitation

The evaluation of cavitation in this section refers to analytical predictions of the type and extent of cavitation on the faces of the propeller blades. Experimental techniques are considered in Section 12. The purpose of these calculations is to determine how severe the cavitation problem is in a given wake field. They will enable the designer to assess the risk of damage to the propeller from erosion and bent trailing edges and to estimate the magnification of propeller-generated hull pressure over the noncavitating case. This report is concerned with the second of these goals, and the next section discusses techniques for estimating these pressure factors.

Most of the recent research on the theoretical prediction of cavitation on propellers has been done in Europe, principally at the Netherlands Ship Model Basin and at the Swedish State Shipbuilding Experimental Tank. Van Oossanen [33] has reported results in late 1977 which show very good agreement between calculated and observed full scale as well as calculated and observed in the cavitation tunnel for the extent of cavitation. The computer program used at NSMB for these predictions is further discussed in Section 11.

Other predictions have been made by Johnsson [43] at the Swedish State Experimental Tank, but they do not seem to agree as well with the observed cavitation or with van Oossanen's predictions.

9. Evaluate Propeller Cavitation Factors

After the extent of propeller cavitation has been determined from Design Block 8, the next step is to estimate the amount cavitation will increase the pressures on the hull. Research into the problem of propeller-induced forces has been ongoing for several decades, but it was only recognized in the past ten years that transient cavitation influences the hull pressures very strongly. In fact, Lewis and Kerwin stated in a recent paper [44]:

> *While extensive work in noncavitating flows was not entirely a wasted effort, it would seem clear now that both analytical and experimental prediction of vibratory forces are completely unrealistic without inclusion of the influence of cavitation. What remains to be seen is whether or not design decisions based upon minimizing noncavitating propeller vibratory excitation are optimum when cavitation is present.*

Reference 45 reported to the 12th International Towing Tank Conference that the amplitudes of fluctuating pressures increased around 100 percent when cavitation was present between the propeller tip and about 0.85 radius. Other investigations of the variations in these pressure fluctuations are given in References 46-48, and the reported cavitation

factors range from 2 to 40. Clearly, with this large variation in the pressures generated between noncavitating and cavitating propellers, it is not a simple task to estimate a reasonable value of the factor.

These factors must be obtained from experimental tests, either full scale or model. The most straightforward approach is to make pressure measurements on a ship with similar lines, wake, and operating conditions. However, for a new design, this approach may not be feasible. Instances in which such measurements were made are described in References 49-50. Model tests must either be conducted in a cavitation tunnel or a depressurized towing tank facility. For tests in the cavitation tunnel, the wake field is simulated either by screens and an afterbody model of the ship, or if the tank is sufficiently large, a complete ship model. These direct test methods will be described in more detail in Block 12.

The evaluation of the propeller cavitation factors proposed in this section is much less precise in that estimates based on previous model experiments are used as a basis. Reference 48 contains results of cavitation tests conducted in the large cavitation tunnel of the Netherlands Ship Model Basin for a number of Wageningen B-series model propellers. Figure 5 shows the geometry of the stern and the location of the pressure pick-ups on the ship's centerline over the propeller. To use Reference 48 for estimating the cavitation factors, the investigator would first need to evaluate the extent of cavitation on the propeller and compare it with figures given in the reference. The corresponding cavitation index σ_n can now be estimated, and knowing the ship's thrust coefficient, K_T, the amplitudes of the first and second harmonic component of the average pressure fluctuations (ΔP_1 and ΔP_2) can be estimated from the tables given in the reference. Figure 6 shows the extent of cavitation on the front and back faces of a Wageningen BB 4-70 propeller at cavitation indices of 2.5 and 5.0 along with the nondimensional pressure fluctuation corresponding to a thrust coefficient of $K_T = 0.075$. The dimensionless pressure variable $Kp_{1,2}$ is defined as

$$Kp_{1,2} = Kp_1 \sin(z\omega t-\phi_1) + Kp_2 \sin(2z\omega t-\phi_2)$$

where

ωt = angular blade position ($\omega t = 0$ for a blade in the vertical, top position)

z = number of propeller blades

$Kp_1 = \dfrac{\Delta P_1}{\rho n^2 D^2}$ (pressure coefficient for first harmonic)

$Kp_2 = \dfrac{\Delta P_2}{\rho n^2 D^2}$ (pressure coefficient for second harmonic)

$\Delta P_1, (\Delta P_2)$ = amplitude of first (second) component of averaged pressure fluctuation

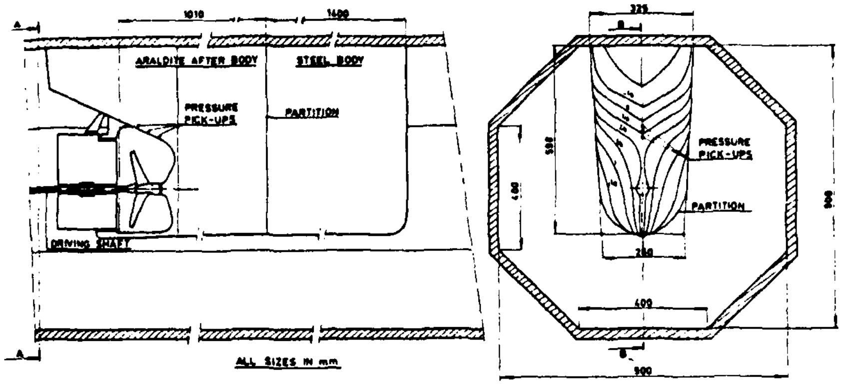
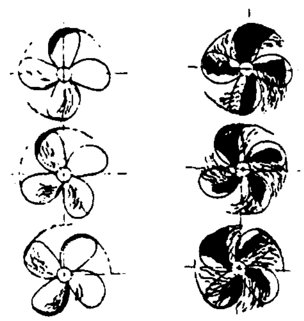
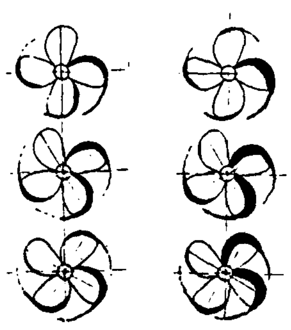


FIGURE 5. STERN GEOMETRY FOR CAVITATION TUNNEL TESTS, FROM [48]

BACK SIDE CAVITATION



FACE SIDE CAVITATION



DIMENSIONLESS PRESSURE FLUCTUATIONS FOR $K_T = 0.075$

First Harmonic Components (Frequency Equal to Blade Frequency)

	place pressure pick-up	K_{P_1}	ϕ_1
$\sigma_n = 5$	forward	0.0184	121
	middle	0.0198	141
(N = 1200 RPM)	aft	0.0135	180
$\sigma_n = 2.5$	forward	0.0462	172
	middle	0.0491	173
(N = 1200 RPM)	aft	0.0395	182

Second Harmonic Components (Frequency Equal to Twice Blade Frequency)

		K_{P_2}	ϕ_2
$\sigma_n = 5$	forward	0.0023	133
	middle	0.0033	135
(N = 1200 RPM)	aft	0.0025	150
$\sigma_n = 2.5$	forward	0.0085	33
	middle	0.0067	12
(N = 1200 RPM)	aft	0.0060	2

FIGURE 6. CAVITATION PROPERTIES OF MODEL SERIES PROPELLER AT $K_T = 0.075$, FROM [48]

ρ = density of water

n = number of propeller revolutions per second

$\phi_1, (\phi_2)$ = phase angles of first (second) harmonic

D = propeller diameter

The cavitation index σ_n and thrust coefficient K_T are defined as

$$\sigma_n = \frac{P_o - P_v}{\frac{1}{2}\rho n^2 D^2}$$

$$K_T = \frac{T}{\rho n^2 D^4}$$

where

P_o = static pressure at the centerline of the propeller shaft at the propeller

P_v = vapor pressure

T = propeller thrust

The success of this procedure clearly depends on how closely the stern configuration, propeller geometry, and wake match the test conditions. The method will not yield exact results, but may provide valuable information as to the magnitudes of the cavitation pressure factors. The results are also limited in that they provide information only at the measured locations and not at other points on the ship's hull.

10. Direct Calculation of Cavitation Pressures and Forces

After calculation of the pressure forces and moments as discussed in Section 6, the next task is to compute the propeller-generated pressures on the hull. These pressures come from three sources as discussed in Section 7. The first results from a fluid element being displaced from one point to another, the second is due to an expanding or contracting fluid volume caused by cavitation, and the third is the pressure due to the propeller loads. As discussed in the previous section, the cavitation-induced pressures can increase the total hull pressure by several factors and play a major role in producing propeller-induced vibrations. Cavitation also reduces the propeller's service life in the form of erosion and bent trailing edges. Consequently, accurate prediction of cavitation would be a major step forward in improving ship design.

In recent years, with the increase in ship size, higher speeds and power, the problem of computing cavitation directly has received increased attention. Most of the work is being done in Europe, principally by van Oossanen at the Netherlands Ship Model Basin. Reference 33 provides an

excellent state-of-the-art review in cavitation prediction techniques. From work recently published by van Oossanen, it appears that his technique gives good correlation with observed cavitation patterns for lightly and moderately loaded propellers. The predicted results are not as satisfactory for a heavily loaded propeller in a very non-uniform wake. Van Oossanen attributes this to lack of knowledge of the change in wake flow due to the working propeller.

Reference 8 indicates that the Netherlands Ship Model Basin has a computer program, designated as CAVANAL, for the computation of cavitation on propeller blades. The input data required for the program are listed in Table 6. Reference 33 gives comparisons of the extent of cavitation as predicted by theory and observed in model tests.

TABLE 6. SUMMARY OF DESIGN BLOCK 10--DIRECT CALCULATION OF CAVITATION PRESSURES AND FORCES

Purpose:	To Determine the Extent of Cavitation and Pressure Distribution on a Propeller
Input:	Propeller Geometry Parameters Propeller Diameter Hub Diameter Number of Blades Expanded Blade Area Ratio Radial Pitch Distribution Radial Distribution of: Distance of Leading Edge to Generator Line Distance of Trailing Edge to Generator Line Radius of Trailing Edge Maximum Camber-Chord Length Ratio Maximum Thickness/Chord-Length Ratio Location of Maximum Camber with Respect to Leading Edge Angle Between Nose-Tail Line and Pitch Line Wake Field at 25 Angular Coordinate Values for Five or More Radii Direction of Rotation Propeller RPM Ship Speed Static Pressure at Centerline of Propeller Fluid Minimum Vapor Pressure Water Temperature Fluid Density
Output:	Pressure Distribution on Propeller Blade Cavitation Index Extent of Cavitation on Propeller Blades
References:	8, 33, 51

Recently, the program CAVANAL has been coupled with another NSMB program which calculates the propeller-induced pressure field on the hull. This last program, called HUFO, is based on the theory developed by Noordzij and described in Reference 51. Van Oossanen reports [8] that since CAVANAL and HUFO have been joined, a relatively large number of successful calculations of hull surface pressures have been carried out. These were performed to optimize propulsion configurations for a given hull, given operational parameters, and a given wake.

11. Conduct Model Tests

The model tests discussed in this section refer to those associated with noncavitating propeller-generated forces and pressures. Experimental tests used to study the effects of cavitation will be discussed in the next section. Tests related to other portions of the ship's design were listed in Section 3, where the wake survey was first discussed. The purpose of conducting these model tests is to measure the propeller, hull, and rudder forces for comparison with the values calculated from the analytical techniques. A favorable comparison will confirm the wake survey and give assurance that the excitation levels actually experienced by the ship will be close to those predicted. If the comparison is not within acceptable limits, then the designer must determine what is causing the discrepancies. Possible sources of errors are in the conduct of the wake survey, the determination of the Fourier component of the wake as required by the programs used to compute propeller forces and pressures, inaccuracies in these programs or associated data, or errors in the conduct of model tests. Whatever the source of error, it is imperative that the differences between analytical and experimental predictions be resolved. There is little sense in making response predictions for the entire ship and its subsystems if one does not have confidence in the accuracy of the applied loadings.

The expense of these tests should only be a small portion of the entire budget allocated to the vibration analysis. In fact, the same model which was made for the wake survey can be used. Only a scale model of the final propeller design need be constructed.

The method for conducting these tests was developed by F. M. Lewis in 1936 [23] for the measurement of vertical and lateral forces plus a longitudinal couple. More current techniques for conducting these tests are described in References 52-53. It should be noted that the model tests for measuring propeller forces require propeller dynamometers which are located aft of the propeller on a separate measuring device. The purpose of this arrangement is, of course, to measure directly the forces and moments exerted by the propeller at its connection with the line shafting. David Taylor Model Basin, Massachusetts Institute of Technology, and the Netherlands Ship Model Basin are three test facilities which have propeller dynamometers.

12. Conduct Cavitation Tests

This report has already emphasized the importance of propeller cavitation on the level of vibratory pressures imposed on the ship's hull. It

also has indicated that the theoretical prediction techniques of cavitation and its effect are still being developed and are not available throughout the world. For these reasons, various research organizations have established experimental facilities to study this problem and assist the shipbuilder to design a ship in which detrimental propeller cavitation effects are minimized. Table 7 lists some of the cavitation test facilities throughout the world.

These tests can be conducted by either one of two experimental techniques: the cavitation tunnel or a facility capable of modeling the free surface effects.

In a cavitation tunnel of limited size, the wake is established by screens or a combination of screens and a model of the ship's afterbody. For larger tunnels such as at the Swedish State Experimental Tank, a complete ship model can be used to establish the wake. References 44, 49, and 54 describe the testing techniques and results of several investigations conducted in cavitation tunnels.

For tests conducted in a variable pressure towing tank, the dynamic and kinematic similarities between the ship model and prototype are matched as closely as possible. The usual procedure is then to maintain equal cavitation and Froude numbers for the model and prototype. The largest depressurized towing tank is operated by the Netherlands Ship Model Basin in Ede, The Netherlands. A description of this facility along with some recent experimental results is presented in Reference 55.

References 56 and 57 describe experimental programs for cavitation studies in both tunnels and variable pressure towing tanks for ducted propellers on large ships with deadweight tonnage over 200,000-dwt. Reference 58 shows that good agreement was obtained when results obtained from the large cavitation tunnel at the Swedish State Experimental Tank were compared with those from the Netherlands Ship Model Basin for a tanker.

13. Compute Total Pressures and Forces

The objective of the previous hydrodynamic investigations was to determine the magnitudes and frequencies of the individual excitations acting on the ship. As indicated in Figure 1, excitations arise from the propeller operating in a given wake field and are transmitted to the ship by three mechanisms: forces and moments exerted by the propeller on the line shafting; pressures, both noncavitating and cavitating, transmitted to the ship hull; and excitations introduced into the ship through hydrodynamic loading of the rudder in the vicinity of the working propeller. All of these have been discussed in the previous sections. The purpose of this design block is to assemble the results of the earlier work and compute the total components of hydrodynamic excitation acting on the ship. These components will then be used as input to determine the forced response for each of the ship's structural subsystems and for the complete ship. The response calculations will be discussed in the next two design phases.

Figure 7 shows a flow diagram for the computation of the total propeller-induced pressures and forces acting on the ship. The reader can

TABLE 7. SOME CAVITATION TEST FACILITIES*

Organization	Location	Type of Facility
Massachusetts Institute of Technology	Cambridge, Massachusetts, USA	Variable Pressure Water Tunnel
David W. Taylor Ship Research and Development Center	Carderock, Maryland, USA	2 Variable Pressure Water Tunnels
Netherlands Ship Model Basin	Ede, The Netherlands	Variable Pressure Towing Tank
Ship Research Institute of Norway	Trondheim, Norway	Cavitation Tunnels (2)
Swedish State Shipbuilding, Experimental Tank	Göteborg, Sweden	Cavitation Tunnels (2)
VWS	West Berlin, Federal Republic of Germany	Free Surface Tunnel

*This list is not necessarily complete.

contribution of cavitating and noncavitating propeller pressures. Refer-
ence 21 presents theoretical and experimental data which show the advantages

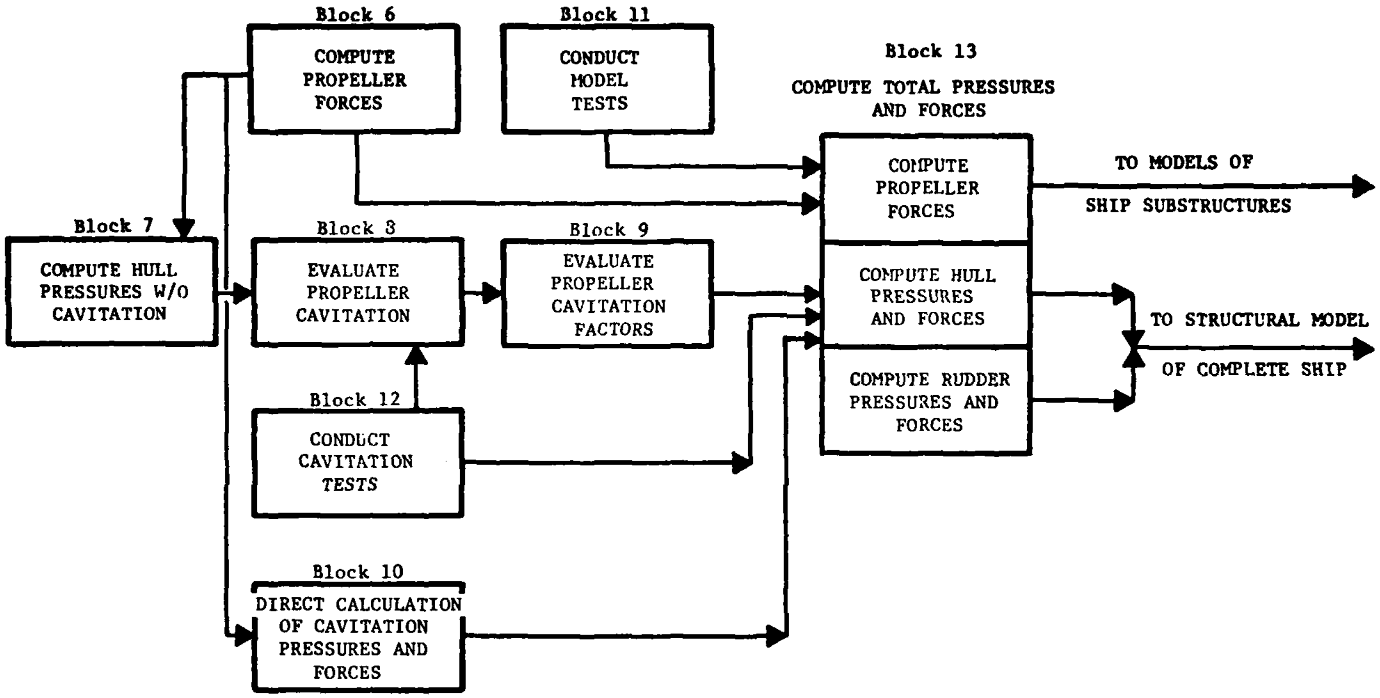


FIGURE 7. FLOW DIAGRAM FOR COMPUTATION OF TOTAL PROPELLER-INDUCED PRESSURES AND FORCES

see that all design blocks in the final hydrodynamic phase (Blocks 6-12) are used in the process. The calculation of the propeller forces and moments is fairly straightforward and was discussed in Section 6 along with available methods. It was pointed out in Section 11 that model tests should also be carried out to confirm the wake survey used in the calculation of these moments and forces. The results of the model tests could supplant the numerical calculation, but this is not recommended. In general, the calculation procedures are much less time consuming and costly than the experimental tests and give a broader picture of the excitation. Therefore, it is probably desirable to carry them through even if detailed model tests are run.

The computation of the hull pressures and forces is more difficult than that for the propeller forces and moments. It is complicated even more by the influence of cavitation, whose importance was only quantitatively recognized within the last ten years. At the present time, the usual way of evaluating the total hull pressures is to compute the pressures without cavitation, estimate the effect of cavitation, and apply appropriate factors to determine the total excitation. These procedures were discussed in Sections 7-9, where it was indicated this process is roundabout and not very satisfactory from a theoretical point of view. A direct calculation of the cavitation pressures and forces is certainly more desirable, and research is active in this area. Section 10 discusses the direct calculation method used by the Netherlands Ship Model Basin.

The other method of obtaining the cavitating hull pressure is through model tests either in a cavitation tunnel or depressurized towing tank. There are several organizations throughout the world which have experimental facilities especially designed for conducting these tests. These were discussed in Section 12 along with references to some recently published results. As in the case of the propeller forces, the cavitation model tests could be used in place of analytical predictions of the extent and pressure generated by cavitation. The word of caution expressed earlier in this section still applies. Model tests are expensive to conduct, and theoretical work needs to be continued so that the phenomenon of cavitation is fully understood and its effect on the ship's design can be analytically predicted.

14. Determine Forced Longitudinal Response of Shafting

By previous calculations, the longitudinal exciting force at the propeller will have been determined. The purpose of this study is to find whether the vibration level generated by this excitation will be acceptable. The propeller and the length and diameter of the shafting will be known, and the thrust bearing will probably have been selected. The unknown quantity will be the stiffness of the thrust bearing foundation. The amplitude of motion at the thrust bearing as a function of frequency for different values of foundation stiffness is required.

The stiffness of the thrust bearing foundation must be determined to assure that a shaft longitudinal vibration resonance does not fall in the operating range. For preliminary analyses, the foundation static stiffness can be determined by the use of several methods. The simplest

process is to represent the foundation and bottom as a combination of frustums of wedges and beams. This procedure is described in Reference 12. A process requiring less engineering judgment is to use finite-element methods, assuming that the machinery double bottom is supported at its edges. It is also possible to represent the machinery space double bottom as an anisotropic plate.

Generally, it will be found that the natural frequency of the bottom structure will not be far removed from the propeller blade frequency. If coincidence occurs, the propeller through longitudinal vibration of the shafting will excite engine room vibration even though the natural frequency of the shaft in longitudinal vibration determined from static stiffness considerations appears to be suitable. This aspect is considered in the following section.

As in Design Step 4, in which the longitudinal propulsion frequencies were estimated, the maximum excitation frequency should be less than about 80 percent of the longitudinal resonance frequency. If this is not possible, then the excitation frequency at the lowest steady operating speed should be about 30 percent above the critical longitudinal frequency.

Several types of computer programs are suitable for this analysis. The system can be broken down to a sequence of masses connected by springs. This can be analyzed by a Holzer Table program, the kind developed for torsional vibration, or by a standard finite element program such as ANSYS, MARC, STARDYNE, NASTRAN, SESAM, etc. However, the shafting, whose distributed weight is several times that of the propeller with its associated water inertia, consists of long lengths of constant diameter. This characteristic is encouraging to a program that represents the shaft as distributed mass and elasticity, and a few computer programs have been developed which utilize this property. In such a case the system can be defined with a minimum of input variables, thus saving time, improving accuracy, and reducing the probability of erroneous inputs.

Table 8 shows the information necessary for conducting this forced vibration analysis. All of the input data will be available either as part of the basic ship design parameters or generated during the design process.

The following appendices taken from Reference 59 indicate that the Maritime Administration has a program (Appendix C-1) based upon the Holzer Method for determining longitudinal vibrations; that J. J. McMullen has a program (Appendix C-2) for determining longitudinal vibration where the shaft is modeled as lumped masses; and that Newport News has a program (Appendix C-3) that can represent the shaft as a distributed mass system. A Littleton Research and Engineering Corp. program utilizing lumped and distributed masses and elasticities is described in Appendix C-4.

15. Determine Forced Response of Machinery Space

The shafting system is connected in longitudinal vibration to the machinery space double bottom through the thrust bearing. Thus, vibrations of the shaft will be coupled with those in the machinery space, and

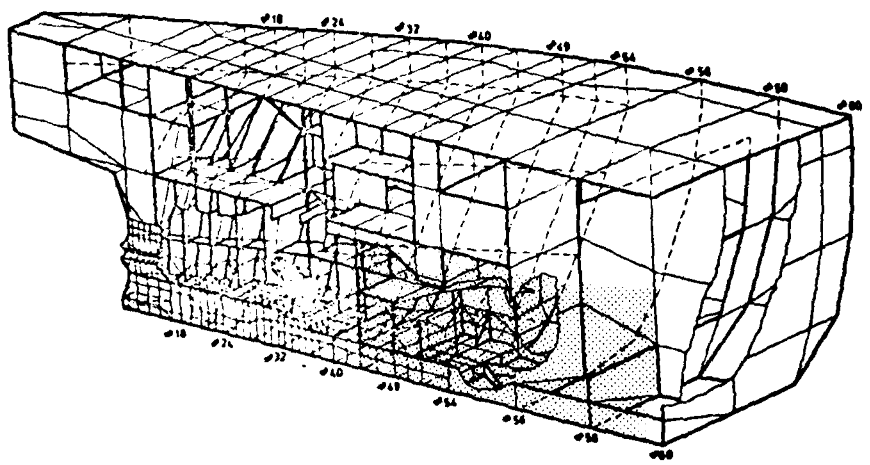
TABLE 8. SUMMARY OF DESIGN BLOCK 14--DETERMINE FORCED LONGITUDINAL RESPONSE OF SHAFTING

Purpose:	To Compute the Resonant Frequencies and Response Amplitudes of the Longitudinal Shafting Subjected to Forced Excitations
Input:	Stiffness of Thrust Bearing Foundation Thrust Bearing Stiffness Reduction Gear Mass and Stiffness Length and Diameter of Shafting Mass and Added Mass of Propeller Estimates of Damping Longitudinal Excitation Levels
Output:	Longitudinal Shafting Resonant Frequencies Amplitude of Motion at Thrust Bearing
References:	12-15, 59

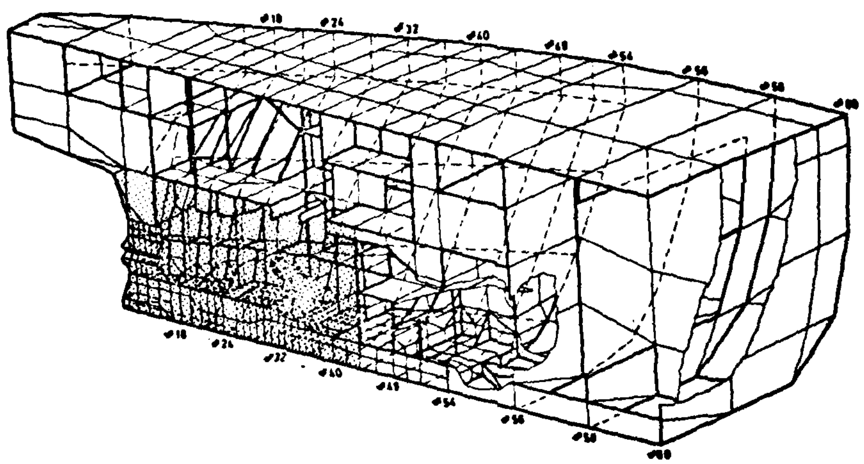
vibrations in the machinery space bottom structure can be strongly coupled with longitudinal vibration of the shafting. Although it might be desirable to model the double bottom as an anisotropic plate with variable inertias for the same reasons that the distributed mass-elasticity procedure is used for the shafting, this type of model has not been developed, and it is necessary to use finite-element modeling. Reference 60 discusses the modeling techniques used in a recent static and dynamic finite-element analysis of the hull structure of a large surface-effect ship. The anisotropic behavior of the hull plating due to longitudinal T-shaped stiffeners was incorporated into the NASTRAN model by modifying the material property matrix.

It is desirable that the computer system that is used be compatible with that used for the complete ship. If the final ship is to be modeled by finite-element procedures, the same system should be used for the machinery space, which can then be incorporated in the full model as a substructure. If the complete ship is to be modeled as a Timoshenko beam with sprung masses, any convenient finite-element model can be used for the machinery space.

Figure 8(a) shows a finite element of the afterbody with the machinery space shaded. The model of the longitudinal shafting and thrust bearing stiffnesses will be available from the analysis conducted in Block 14. The question which must be decided is to what extent and in what detail must the surrounding machinery space structure be modeled to represent adequately its mass and stiffness properties. The answer depends a great deal on the experience of the engineer conducting the analysis. The model should extend in the transverse direction over the ship's half width (symmetry can be assumed), in the longitudinal direction to at least several



(a) Machinery space is shaded



(b) Aft bearing structure is shaded

FIGURE 8. FINITE ELEMENT MODEL OF A SHIP'S AFTERBODY, FROM [61]

transverse floors on each side of the thrust bearing, and preferably the whole length of the machinery space, and in the vertical direction from the bottom of the ship's hull to about two decks above the thrust bearing. The important point to remember is that the model must be accurate enough to predict the lower natural frequencies and mode shapes.

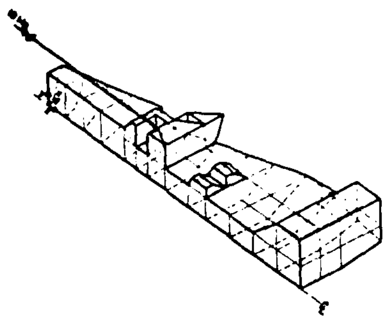
If doubts exist in choosing a proper model size, the analyst should err on the conservative side; i.e., pick a model larger than necessary. First, the computed resonant frequencies and response will be closer to the actual values. Second, the additional work required to generate the larger model is not lost because the model will be used as a substructure in the complete ship vibration analysis.

One of the earliest finite-element investigations into the natural frequencies of the machinery space was made by Reed [62]. Other investigators, especially in Europe, are using finite-element techniques on a routine basis for the structural analysis of all critical locations in the ship. G. C. Volcy at Bureau Veritas has been particularly active in this area, and many techniques and results obtained by that organization in its study of propeller-induced vibrations are published together in Reference 63. For particular papers dealing with machinery space vibrations, the reader should see References 64-66.

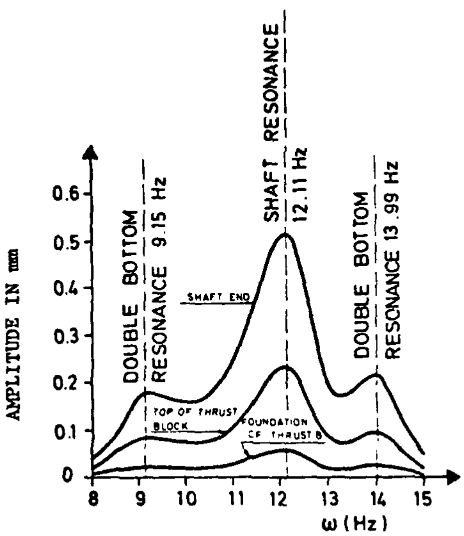
Figure 9(a) taken from Reference 67 shows a finite-element model of the shaft and double bottom for a large tanker. Table 9 gives the natural frequencies for the separated and integrated models and indicates the double bottom affects the shaft's frequency more strongly than the shaft affects the double bottom's frequency. Figure 9(b) clearly indicates a shaft resonance at 12.11 Hertz, and the corresponding mode shape is given in Figure 9(c). Reference 67 indicated that the static stiffness of the thrust block was 0.190×10^6 Mp/m (one Mp equals 1000 kilograms of force), which resulted in a first mode longitudinal shaft frequency of 12.83 Hertz. When the double bottom model was coupled with the shaft, the frequency

TABLE 9. NATURAL FREQUENCIES OF DOUBLE BOTTOM AND SHAFT FOR SEPARATED AND INTEGRATED MODELS, FROM [67]

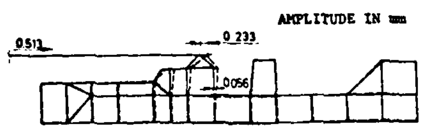
	Double Bottom	Shaft	Double Bottom and Shaft
1st mode of double bottom	9.20 Hz		9.15 Hz
1st mode of shaft		12.83 Hz	12.11 Hz
2nd mode of double bottom	14.06 Hz		13.99 Hz



(a) Double bottom and shaft model of 172,000 dwt tanker



(b) Resonant response of the center girder of 172,000 dwt tanker, ω = 12.110 Hz, thrust force of 10 Mp



(c) Response of shaft and thrust block of 172,000 dwt tanker, thrust force 10 Mp at ω = 12.110 Hz

FIGURE 9. FINITE ELEMENT MODEL AND RESPONSE OF THE DOUBLE BOTTOM AND SHAFT FOR A 172,000 DWT TANKER, FROM [67]

decreased to 12.11 Hertz, and Figure 10 shows that this coupling decreases the effective stiffness of the thrust block foundation by approximately 13 percent. These results indicate that a separated shaft model can only be used in a vibration analysis where the natural frequencies are not close to double bottom resonances.

Table 10 shows the information pertinent to the conduct of this analysis. The only input data in addition to that required for the forced response of Section 14 are estimates of the machinery space mass distribution and the ship's scantling plans.

16. Determine Forced Lateral Response of Shafting (Rigid Hull)

The shaft responds laterally to the harmonic force and moment excitations about axes normal to the rotational axis. If the lateral natural frequencies of the propeller and shaft system coincide with the blade frequency excitation, the input to the hull through the bearings can be strongly amplified. Calculations of ship response generally show peaks associated with lateral frequencies of the shafting. It is, therefore, desirable to design the shafting system so that these resonances will not occur at the normal operating speeds. As with the longitudinal vibrations, these studies are successively made on models of increasing complexity. The first studies are applied to the shaft simply supported at the bearings (either at the forward and after edges or one-third of the distance from the rear of the stern bearing). Since it is known that the bearings are relatively flexible, this model will generally give a frequency that is high so that if the lowest lateral frequency is less than, say, 30 percent above the full power blade frequency, it will probably be wise to consider relocating the bearings or modifying the shafting to raise the frequency. Bureau Veritas in its Guidance Note [68] published in 1971 recommends, for example, that the natural frequency be above 130 percent of the excitation frequency.

It is also important to ensure that all vertical bearing reactions be positive, i.e., maintain contact with its support, over the range of excitation. A negative reaction will result in hammering of the bearings. This hammering is certainly undesirable from the standpoint of maintenance and noise levels. However, it also changes the system's natural frequency and could move it closer to the propeller-excitation frequencies. Large lateral vibratory motions of the shaft would result, and these vibrations would probably be transmitted to other portions of the ship. References 69 and 70 present results of investigations conducted at Bureau Veritas on lateral shafting vibrations and the associated wearing on the bearings caused by misalignment of the propeller shaft.

To determine the forced vibration response, the analyst can use any of the commercially available computer programs such as ANSYS, ASKA, NASTRAN, SESAM, STARDYNE, and STRUDL. There is no optimum choice among these programs because all are highly developed and give essentially the same type of results. The choice depends on the availability of the program and the familiarity of the user with it. A discussion of the theoretical aspects of the finite-element method as it pertains to transverse vibrations of a ship's propulsion system is given in Reference 71.

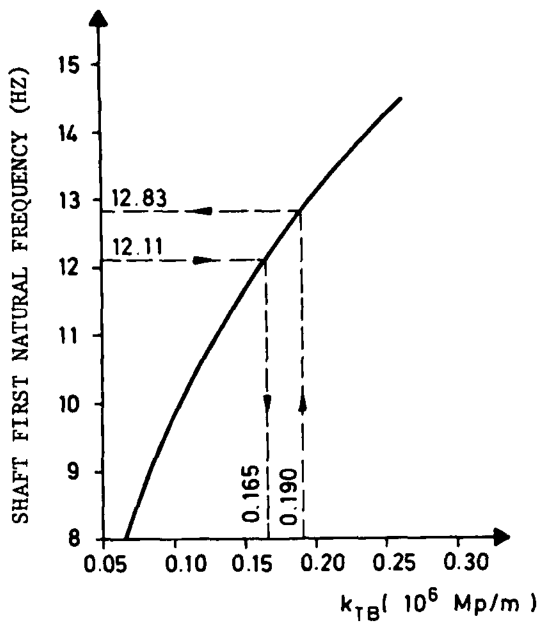


FIGURE 10. DEPENDENCE OF THE FIRST SHAFT NATURAL FREQUENCY ON THRUST BLOCK STIFFNESS, K_{TB}, 172,000 DWT TANKER, FROM [67]

TABLE 10. SUMMARY OF DESIGN BLOCK 15--DETERMINE FORCED RESPONSE OF MACHINERY SPACE

Purpose:	To Compute the Resonant Frequencies and Response Amplitudes of the Longitudinal Shafting Including the Mass and Stiffness of the Machinery Space
Input:	Structural Plans Machinery Space Mass Distribution Thrust Bearing Stiffness Reduction Gear Mass and Stiffness Length and Diameter of Shafting Mass and Added Mass of Propeller Estimates of Damping Longitudinal Excitation Levels
Output:	Longitudinal Shafting Resonant Frequencies Amplitude of Motion at Thrust Bearing
References:	60 - 67

Two other methods of determining the transverse response should be mentioned at this point. These are the transfer-matrix and finite-difference methods. Reference 72 discusses the transfer-matrix approach and applies a particular computer program to the analysis of several ships. The finite-difference method is also used in a fully operational computer code available from the David W. Taylor Naval Ship Research and Development Center. This program, called GBRP [73], can also treat longitudinal and torsional vibrations of the shafting as well as the coupled lateral and torsional vibration of a beam. This makes it particularly well suited to study the bending response of a ship which can be idealized as a beam. This application of the GBRP program will be discussed later in the report.

Since the analysis described in the next section is an extension of the one assuming a rigid hull, it is wise to choose a computer program which can be used for both. This will allow the analyst to use the already existing model of the shafting and only add the flexibility and mass of the surrounding hull structure. With the ability of general purpose computer programs to model longitudinal, lateral, and torsional vibration of beams with the same finite elements, it may be possible for the analyst to use the model already developed for the longitudinal vibration of the shaft. Table 11 shows the information necessary to carry out this analysis of the shafting with rigid lateral supports.

TABLE 11. SUMMARY OF DESIGN BLOCK 16--DETERMINE FORCED LATERAL RESPONSE OF SHAFTING (RIGID HULL)

Purpose:	To Compute the Resonant Frequencies and Forced Response of the Lateral Shafting Along with Shaft Bearing Reactions
Input:	Length and Diameter of Shafting Mass and Added Mass of Propeller Location of Bearings Oil Film Stiffness Estimates of Damping Vertical and Horizontal Excitation Levels for Propeller Forces and Moments
Output:	Resonant Frequencies Response Amplitudes of Shafting Versus Excitation Frequency Static and Dynamic Bearing Reactions
References:	68 - 73

17. Determine Forced Lateral Response of Shafting (Flexible Hull)

The analysis conducted in Design Block 16 will ensure the response of the shafting to lateral excitation will be acceptable if the bearings are assumed rigid. This is the simplest type of analysis, and the next level of sophistication assumes some flexibility in the bearing supports. The information produced by this investigation is again the resonant frequencies and forced response of the lateral shaft along with the bearing reactions. The dynamic bearing reactions should again be positive to avoid slamming and gradual loss of contact.

If gyroscopic effects are neglected (they are important for whirling, but relatively unimportant at blade frequencies), and the supports are of equal stiffness in all directions, the natural frequencies and response of the shaft will be the same in all directions. If the structure is not symmetrical, the fundamental normal modes may be in any pair of orthogonal planes. The moment restraint at the bearings can also have a significant influence on the shaft frequency.

The amount of structure to include in the stiffness calculation is a matter for the analyst's judgment. The object is to evaluate the stiffness to a region of a large hull mass. For a single-screw ship, this may involve the structure from the after peak bulkhead and up to the steering gear flat. For shafts supported by struts, it will include the struts and their backup structure. If the complete hull is analyzed using a finite-element analysis, the validity of the modeling can be tested.

Since the structure supporting the shaft bearings is complicated, the use of finite-element methods is the most feasible way of determining the support stiffness. Figure 8(b) is a finite-element model of the afterbody of a ship with the aft bearing structure shaded. When this model is generated, consideration should be given to the fact that it will also be used as a substructure in the complete ship analysis. The basic philosophy is to have adequate representations of each subsystem available so they can be assembled as efficiently as possible. The stiffness between the shaft and bearing of a stave bearing can be quite low if the staves are rubber. The stiffness of an oil film bearing is such that a bearing force introduces a motion having a component perpendicular to the load.

If, as a result of the calculations of shaft response, it is found that there are no shaft resonances within about 30 percent of the operating speeds of the ship, the shafting can be considered satisfactory for this level of refinement. Later analyses of the whole ship will confirm its suitability. If, on the other hand, lateral resonances appear close to the operating speed, then by changing one or more of the following, a new propulsion system can be developed which has resonances properly located:

1. The overhang of the propeller beyond the stern bearing.

2. The span between the last two bearings supporting the propeller shaft.

3. The diameter of the propeller shaft.

4. The support of the propeller shaft bearing:

   a. The skeg and stern tube structure for a single-screw ship having a skeg supported bearing.

   b. The angles, size, attachment to the bearing barrel of the arms carrying a strut bearing for open-screw ships.

   c. The structure supporting strut arms.

   d. Other changes as indicated by the calculations.

Such changes are frequently required, and good judgment, often using analyses of simple models, is required to discover the optimum solution rapidly and inexpensively.

The references listed in the last section are all applicable to the shafting analysis considering flexible lateral supports. In addition, Reference 74 addresses specifically the problem of shaft vibrations in elastically supported tail shafts.

Table 12 summarizes the information required to conduct this analysis. With the exception of the scantling plans necessary to model the mass and stiffness of the hull structure surrounding the shaft, no input data in addition to that used in Table 11 are required. Appendix D contains several computer programs specifically written for the analysis of lateral vibration of the shafting. However, any of the commercially available finite-element computer codes could also be used.

18. Conduct Superstructure Modal Analysis

In addition to substructures of the shafting and machinery spaces, it is desirable to make a study of the superstructure as a subsystem since resonances in this region are a frequent cause of vibration troubles. Finite-element methods are generally most suitable for modeling this structure. With the high superstructures common on container ships and very long ships, some superstructures vibrate fore and aft as a cantilever beam. On others the decks vibrate symmetrically within the sides, while on still others the decks vibrate anti-symmetrically (port up, starboard down) so that the finite-element model should not be too coarse to suitably represent the complexity of possible modes.

Figure 11 taken from Reference 75 shows three levels of sophistication for the superstructure models. Generally, the beam and two-dimensional models do not represent adequately the structure, and a three-dimensional model is required. One of the most important considerations in the superstructure analysis is how it is connected to the rest of the ship. This is shown schematically by the springs in Figure 11. As a first approximation, the analyst could assume fixed and simply-supported conditions and determine their effect on the natural frequencies and mode shapes. A better approach is to include in the model a portion of the ship extending

TABLE 12. SUMMARY OF DESIGN BLOCK 17—DETERMINE FORCED LATERAL RESPONSE OF SHAFTING (FLEXIBLE HULL)

Purpose:	To Compute the Resonant Frequencies and Forced Response of the Lateral Shafting Along with Shaft Bearing Reactions when Flexibility of the Surrounding Hull Structure is Considered
Input:	Length and Diameter of Shafting Mass and Added Mass of Propeller Location of Bearings Oil Film Stiffness Estimates of Damping Hull Scantling Plans Vertical and Horizontal Excitation Levels for Propeller Forces and Moments
Output:	Resonant Frequencies Response Amplitudes of Shafting Versus Excitation Frequency Static and Dynamic Bearing Reactions
References:	68 - 74

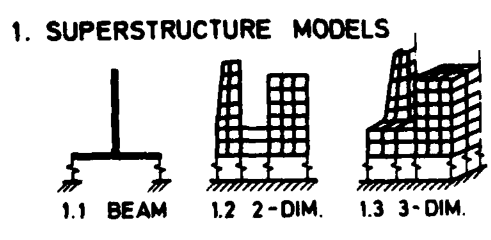


FIGURE 11. THREE LEVELS OF SUPERSTRUCTURE MATHEMATICAL MODELS, FROM [75]

below the main deck. This is illustrated in Table 13 [76] which shows calculated and measured fundamental natural frequencies of the superstructure as a function of the type of finite-element model. In this case, the model containing the engine-room below the superstructure has an eight percent error with the measured value. The error, of course, decreases as more of the ship is included. Whichever type of model the analyst chooses, he should realize that the representation will be used as a substructure for the complete ship analysis.

On ships whose superstructures vibrate badly, it has been found that the natural frequencies generally are in resonance with the propeller excitation frequencies. For this reason, it is recommended that these frequencies should not be within about 25 percent of each other. If this condition is not met, then modifications must be made to the superstructure or its surrounding foundation. Figure 12 shows the mode shape at 10.5 Hertz and indicates that the superstructure vibrates essentially as a rigid body on a flexible foundation. If the blade rate frequency of the propeller is 10 Hertz, then the superstructure should probably be stiffened at the points shown to raise its fundamental frequency to about 12.5 Hertz. The adequacy of these modifications can be confirmed when the analysis of the complete ship is conducted.

Table 14 shows the information necessary to conduct the superstructure analysis.

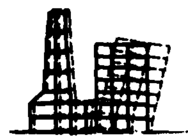
TABLE 13. CORRELATION BETWEEN MEASURED AND CALCULATED SUPERSTRUCTURE FUNDAMENTAL RESONANT FREQUENCY FOR DIFFERENT FINITE-ELEMENT MODELS. 138,000 DWT TANKER, BALLAST CONDITION, FROM [76]*

Finite Element Model		Model Size, Number of				Frequency	Error
		Elements	Nodes-	Supernodes	DOF	Hz	%
A		464	130	52	61	11.5	39
B		1190	400	68	165	9.0	8
C		1500	510	134	315	8.8	6
D		2150	810	185	445	8.2	-1

A - Superstructure clamped at main deck
B - Engine-room below superstructure included (mass free)
C - Complete model of afterbody
D - Total ship included (only starboard side is considered in all models)

*Superstructure fundamental frequency measured value 8.3 Hz

SHAFT RPM = 100
NO. BLADES = 6
BLADE RATE
FREQUENCY = 10 Hz



LOWEST FREQUENCY
f = 10.5 Hz

FIGURE 12. FIRST-MODE SHAPE OF THREE-DIMENSIONAL SUPERSTRUCTURE MODEL, FROM [75]

TABLE 14. SUMMARY OF DESIGN BLOCK 18--CONDUCT SUPERSTRUCTURE MODAL ANALYSIS

Purpose:	To Determine How Close Natural Frequencies of Free Vibration of the Superstructure Are to the Harmonic of Blade Rate Frequencies
Input:	Scantling Plans Estimates of Structural and Equipment Mass
Output:	Superstructure Natural Frequencies Superstructure Mode Shapes
References:	75 - 77

19. Determine Natural Frequencies and Forced Response of Rudder

Figure 1 indicates that excitations from the propeller can enter the ship through the rudder as well as through the propulsion shafting and hull. Although experience has shown that these rudder excitations are not generally as severe as the latter two, care must still be taken in the design stage to avoid them. This can best be done by designing a rudder whose natural frequencies are not close to the propeller's blade rate frequency or its harmonics and which will be subjected to accepted levels of propeller-generated pressures. Procedures for calculating the natural frequencies of the rudder are further developed than those for calculating the excitations.

A ship's rudder is essentially a flat plate immersed in a fluid. In order to compute its natural frequencies, the stiffness and mass of the system must be known. The system's stiffness is simply the stiffness of the rudder with no surrounding fluid. The mass, however, is composed of two quantities: the structural mass of the rudder and the so-called added mass of the surrounding fluid. This added-mass effect reduces the natural frequencies of the structure from their in-air values.

There have been many investigations to calculate the added masses of different geometric shapes. For example, the added mass of a thin rectangular plate was computed in Reference 78 using potential theory, and experimental tests on the vibration characteristics of cantilever plates in water were conducted in Reference 79. With the development of the finite-element technique, numerical methods are now available for computing on a routine basis the free vibration natural frequencies and mode shapes of an arbitrarily submerged structure. The results of one such recent investigation on partially and fully submerged cantilever plates are reported in Reference 80. The numerical calculations were made using the NASTRAN computer code, with the plate being modeled by standard plate finite elements and the fluid by isoparametric three-dimensional solid elements. Excellent comparison between numerical and experimental results were obtained, and it appears this finite-element technique can be used routinely on free vibration analyses of ship appendages such as rudders.

The computation of the excitation forces on the rudder is not as simple as the calculation of the natural frequencies. A numerical procedure and computer program using unsteady-lifting-surface theory has been developed at Davidson Laboratory [81] and subsequently modified to take into account the effects of loading and thickness [82]. However, it does not appear that this procedure is used at the present time for design calculations.

There have also been experimental investigations of the interaction between the rudder and propeller operating in a given wake field. The major contributions in this field were made by Professor Frank M. Lewis at MIT. References 52 and 53 indicate that the propeller-generated forces are extremely sensitive to the axial clearance between the rudder and propeller, and by proper placement of the rudder, low values of horizontal force can be obtained. However, the rudder in the experimental tests is essentially rigid, while the full-scale rudder is an elastic body. This elasticity can play an important role in determining the response, and the correct scaling parameters should be included in the model tests.

Currently, little attention is paid to propeller-rudder interaction effects, and clearances are usually chosen on the basis of those recommended in various classification rules. Additional research is required in this area to fully understand this area of propeller-induced vibrations.

20. Evaluate Local Plating Design

Up to this point, the purpose of the proposed design procedure has been to be able to compute the excitation and response of the major components of the ship. The overall objective will be realized in Design Block 22, when the response of the entire ship will be calculated. Even though the overall vibration characteristics of the ship's hull girder and substructure may be acceptable, the possibility still exists that local vibration problems may arise. Reference 7 defines local vibration as the dynamic response of a structural element, deck, bulkhead, machinery, or equipment elements which is significantly greater than that of the hull girder at that location. However, hull girder vibration does not mean much when the nodal length is of the order of the beam of a ship. The concept

of local vibration in this report is more restrictive in that local vibrations are defined in terms of major structural elements rather than the hull girder.

Since local vibration is simply an extension of the global vibration problem, the finite-element techniques proposed in the earlier design blocks would apply. This is not recommended for the entire ship, however, because the analysis would become too complicated and expensive. A simpler approach is taken in which the local plating natural frequencies are removed from the excitation frequency or its harmonics. The response of the plating is not computed. This approach allows the use of analytical expressions for plate vibrations such as found in Reference 83. Reference 68, published by Bureau Veritas, contains formulas for the fundamental natural frequencies of plate panels stiffened in one or two orthogonal directions with various boundary conditions. The effects of the fluid added mass on the vibration of plating adjacent to a fluid boundary are also considered in Reference 68. These formulas are generally not easy to apply because there are difficulties in defining the extent of the structure taking part in the vibration, the geometric boundary conditions, and the structural mass. In critical areas of the ship where the structure is complicated, these frequencies can be computed using the finite-element method. Also, added-mass effects of external or internal liquid can be included by the techniques explained in Reference 80.

Generally speaking, it would be too complicated to compute the natural frequencies of all local plating in the ship. By using an experienced senior draftsman or designer, most of the structural details of the local plating can be finalized in accordance with standard practice and experience. Only when a new structural arrangement or an unusually large unsupported area is proposed, is a more detailed analysis warranted. If local plating vibrations do occur, corrective measures are relatively easy to implement before the ship's finish work is completed. These local vibrations can be found during the shaker tests which will be discussed in Design Block 23.

TABLE 15. SUMMARY OF DESIGN BLOCK 20--
DESIGN LOCAL PLATING

Purpose:	To Determine How Close Natural Frequencies of Free Vibration of the Local Plating Are to the Harmonics of the Propeller-Blade Rate Frequency
Input:	Scantling Plans Equipment Mass Estimates Added Mass Estimates of Adjacent Fluid
Output:	Local Plating Natural Frequencies
References:	68, 80, 83

21. Assemble Model of Entire Ship

When the subsystems discussed in Blocks 14-20 have been designed so that it is expected that they will be free of vibration resonances, it is time to make a vibration analysis of the complete ship. This analysis of the full ship fulfills two important functions:

1. It checks and confirms the validity of the boundaries assumed for the substructures.

2. By modeling the ship as a whole, it is possible, with the proper damping, to predict the vibration levels in all parts of the ship as a function of frequency. Comparing these predictions with established acceptable levels allows an assessment of acceptability of the ship at a point in construction where corrections and changes to overcome serious difficulties can be determined and incorporated in the design.

Until approximately ten years ago, the standard technique for the complete ship analysis was to model the ship as a beam structure using a Timoshenko beam as a base. This approach is still satisfactory for certain type ships which are compartmentalized and respond as beam-like structures. Submarines and destroyers fall into this class. However, for ships with wide beams and open deck structures, the models do not give satisfactory results because the hull-girder modes and frequencies are affected by vibrations within these areas. Large containerized and bulk carriers are examples of such ships, and for these cases the finite-element method should be used to define the structure accurately. Even in the cases where beam models can be used, the results are generally not accurate for above the fifth or sixth natural frequency because of shear lag and local vibration effects.

Reference 84 presents a review of the theory for bending, longitudinal, and torsional vibrations of a beam and applies them directly to the ship vibration problem. A more recent review of all analytical and numerical methods used to calculate hull vibration is presented in Reference 85.

As an example of a ship modeled in terms of a beam, consider Figure 13, an uncoupled vertical vibration model, and Figure 14, a coupled lateral-torsional vibration model. The sprung masses shown in these figures represent the various subsystems in the ship. For the analysis of structures as complex as shown in Figures 13 and 14, the computer program GBRP (General Bending Response Program) developed at the Naval Ship Research and Development Center [73] can be used. In the analysis of the ship hull, it can treat vertical as well as coupled lateral-torsional vibration. These capabilities and other features of the program are described in Appendix E-1.

To define the elastic properties of the structure, it is necessary to define the cross-sectional elastic properties. A program and procedure for computing

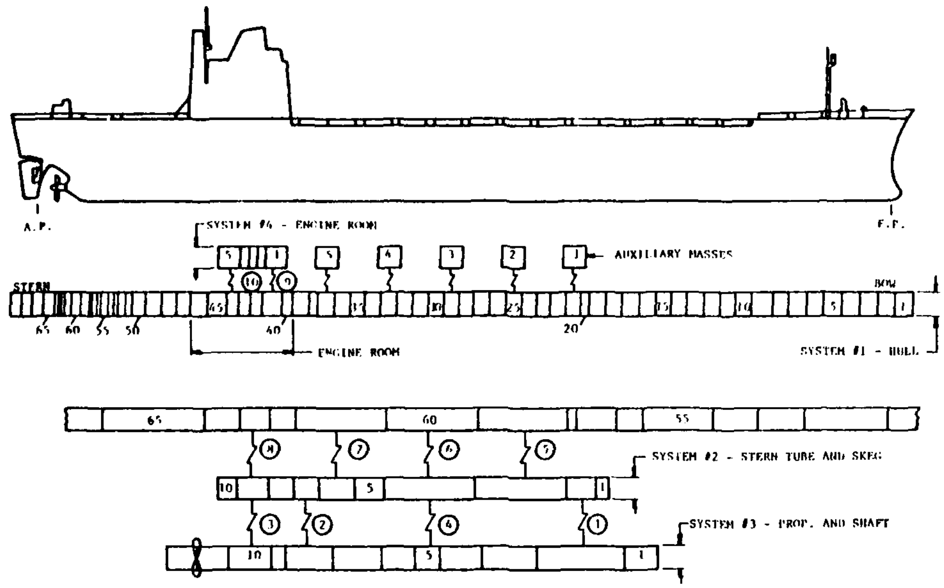


FIGURE 13. UNITIZED AND CONTAINERIZED SHIP, VERTICAL VIBRATION MODEL

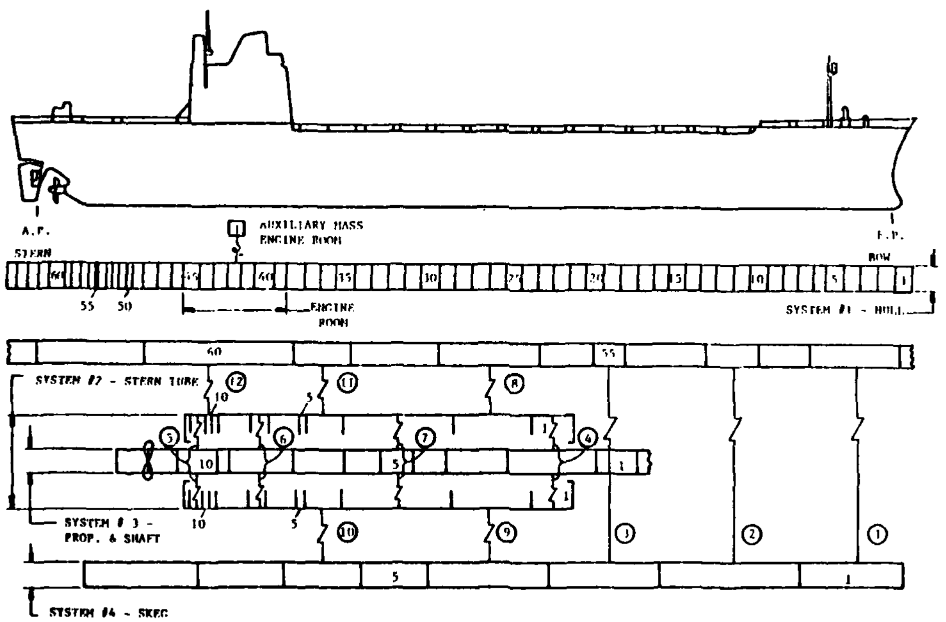


FIGURE 14. UNITIZED AND CONTAINERIZED SHIP, TRANSVERSE VIBRATION MODEL

I_y (Moment of inertia about transverse axis)

I_z (Moment of inertia about vertical axis)

I_{yz} (Product of inertia relative to horizontal and vertical axes)

A (Cross-sectional area)

$K_{xz}A$ (Shear area constant vertical plane)

$K_{xy}A$ (Shear area constant transverse plane)

J_x (Torsional area constant about longitudinal axis)

$\bar{z}$ (Vertical coordinate of the neutral axis)

$\bar{y}$ (Transverse coordinate of neutral axis)

y',z' (Coordinates of the shear center of the section)

are given in Reference 86. This program calculates the equivalent beam parameters for the ship section properties using data tabulations obtained from hull plans by a preestablished orderly procedure.

U. S. Steel Engineers and Consultants, Inc., developed a program that represents a ship as a beam on an elastic foundation. Information on this program is given in Appendix E-2. Other vibration programs based upon modeling the hull as beams have been developed by Lloyd's Registry of Shipping and by the Institut für Schiffstechnik in Berlin, Federal Republic of Germany. These programs are described briefly in Appendix E-3.

The advantage of representing a ship by a beam model is that the computer analysis is more direct and more easily interpreted and is considerably less expensive than that with a finite-element analysis for a structure that is as well defined. As mentioned earlier, the disadvantages are that for many ship vibration problems, particularly where the decks are open so that the vibration across the width of the ship is important, the beam representation of the ship is inadequate and a finite-element process is required for satisfactory modeling.

The use of finite-element methods for predicting ship vibrations is becoming widespread. References 63, 75, 77, 87, and 88 discuss the method and give specific examples of its application to ship structures. Finite-element applications in vibrations and structural dynamics problems of ships and other marine structures are discussed in Reference 89, and an extensive bibliography is presented.

Figures 15 and 16 show elevation and isometric views of a finite-element model for a cargo ship. The forward portion of the vessel is represented as a beam with offset masses. This is an acceptable practice in this case because detailed knowledge of the vibration response in the forward area is not required. Excessive vibrations in the habitable and

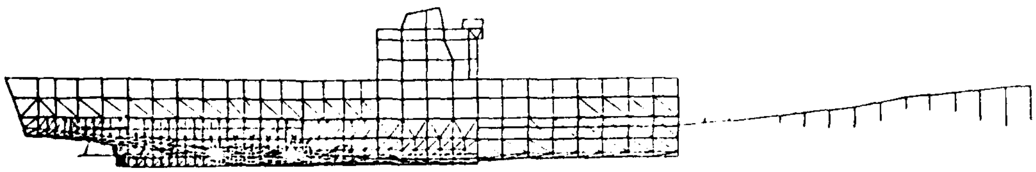


FIGURE 15. ELEVATION VIEW OF FINITE-ELEMENT MODEL

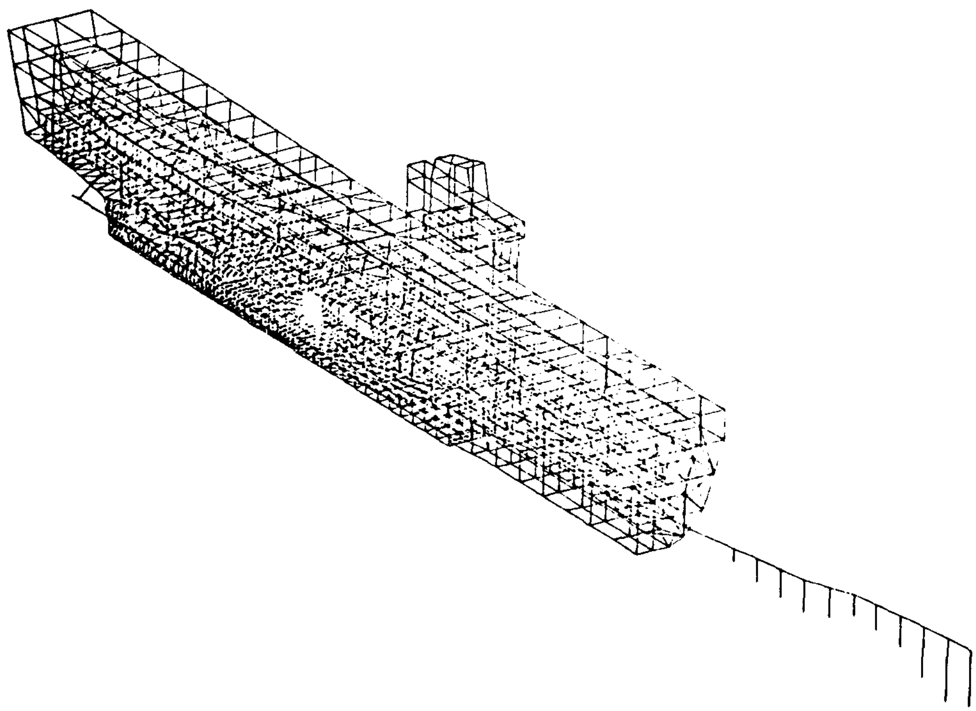


FIGURE 16. ISOMETRIC VIEW OF FINITE-ELEMENT MODEL

and machinery areas are of greater concern, and Figures 15 and 16 show those portions of the ship are modeled in great detail.

Some organizations have developed finite-element programs specifically for ship applications. Among these should be mentioned SESAM-69 (Appendix E-3) developed by Det norske Veritas, the Norwegian classification society [90], and DASH (Appendix E-3) developed by the Netherlands Ship Research Center. Bureau Veritas, the French classification society, also has made many finite element analyses of ship structures. The Electric Boat Division of General Dynamics Corporation developed and maintains a finite-element computer program GENSAM for use on submarine vibration problems. Information on this is presented in Appendix E-3.

Since the cost of developing, maintaining, and updating a large finite element computer program is high, it is common to apply general purpose computer programs to ship vibration problems. In particular, NASTRAN is used by Lloyd's, the American Bureau of Shipping, Littleton Research and Engineering Corp., and probably by other organizations. This program was developed for the analysis of large structures and readily applies to ships. It is being updated continually to improve the representation of structural elements and the processing efficiency. Because it was developed for large projects, NASTRAN carries high overhead structure. Most ship finite-element studies involve models that are large enough to benefit from the generality provided by the overhead, but for many small studies that will not later be incorporated in the large model, it may be desirable to use other finite element programs such as STARDYNE, ANSYS, MARC, STRUDL, or SAP. Again, the choice should be dictated by the program's availability and experience of the user.

This section has discussed the various ways of modeling the entire ship and has presented computer programs which are used for this purpose. A summary of this design block is given in Table 16. This model will be used in the next design step to predict the vibration levels throughout the ship.

22. Determine Vibration Amplitudes and Stress Levels of Complete Ship

At this point in the design process, all the ingredients necessary to compute the forced vibration response of the entire ship are available. The hydrodynamic excitations are available through the methods discussed in Design Blocks 6-13, and a mathematical model of the entire ship structure can be assembled in Design Block 21. This step in the design process simply applies the excitations to the model and determines the vibration response. Table 17 provides a summary of this design block and indicates the only information not yet discussed is the estimate of added mass and damping.

The concept of added mass as it applies to the rudder was discussed in Design Block 19. For the complete ship, it is generally assumed the added mass underway is the same as for the ship at rest. At the present time, there is little known about the differences.

Damping is, of course, very important in response calculations because without damping, the resonant amplitudes of vibration have infinite values.

TABLE 16. SUMMARY OF DESIGN BLOCK 21--
ASSEMBLE MODEL OF ENTIRE SHIP

Purpose:	To Construct a Mathematical Model of the Entire Ship Which Can Be Used to Predict the Response to Propeller-Induced Excitations
Input:	Substructure Models of Superstructure, Machinery Space, Stern Structure, and Propulsion Shafting Scantling Plans Mass Distribution
Output:	A Mathematical Model of the Entire Ship
References:	63, 75, 77, 84-90

TABLE 17. SUMMARY OF DESIGN BLOCK 22--DETERMINE
VIBRATION AND STRESS LEVELS OF ENTIRE SHIP

Purpose:	To Compute the Forced Vibration Amplitudes and Stress Levels of the Complete Ship Under Propeller-Induced Excitations
Input:	Mathematical Model of Entire Ship Computed or Measured Propeller Forces and Moments Computed or Measured Hull Forces or Pressures Computed or Measured Rudder Forces Locations on Ship Where Vibration Responses Are Desired Estimates of Damping Estimates of Ship's Added Mass
Output:	Plots of Vibration Amplitude Versus Propeller RPM at Selected Locations Plots of Maximum Stress Versus Propeller RPM at Selected Locations Resonant Frequencies at Selected Locations
References:	63, 75, 77, 84-93

Investigations have shown that damping in ships is dependent on the vibration mode, the manner of construction and type of construction material, and the type of cargo. It can be divided in the three forms

- Structural, or hysteretic, damping
- Coulomb, or dry friction, damping
- Viscous, or fluid, damping,

and all three types are present in ship structures. Damping is usually calculated on the basis of logarithmic decrement measurements obtained either in vibration generator or anchor drop tests. Kumai [91] addressed the question of how damping is affected by the higher modes of vibration, and in Reference 92 values of the hysteretic damping coefficients, g, are given for several United States, Japanese, and English ships in the vertical modes of vibration. A value of $g = 0.029$ was recommended to be used in the General Bending Response Program, at least until further data could be obtained. However, each organization conducting analyses of this type has its own recommended values of damping.

Specific results obtained from a particular forced vibration analysis will be illustrated in detail in Chapter V when the application of the recommended procedure is presented. At this point it is only necessary to illustrate generally what type of results can be expected. Figure 17 shows a two-dimensional finite element model taken from Reference 75 for a 370,000 dwt tanker. The forward portion of the ship is not represented by a beam in this model. Let us assume that the forced response in the horizontal direction at the top of the superstructure is desired for a vertical force acting at the ship's stern. The assumed viscous damping coefficient was taken to be 3.5 percent of critical damping. Figure 18 shows, however, that the coefficient is an important factor only in the vicinity of the resonant peaks. The analysis presented in Reference 75 shows that the top of the superstructure has two resonant peaks at about 6.7 and 11.0 Hertz, as illustrated in Figure 19. The higher peak is due to the natural frequency of the superstructure. Figure 20 reveals the lower peak is essentially a rigid-body motion of the superstructure induced by a hull bending mode. At 6.67 Hertz a nodal point is located at the aft position of the superstructure which will give extreme rigid-body motion. This is the blade-rate frequency for a five-bladed propeller at 80 RPM service speed. If a six-blade propeller is used, this frequency increases to 8.00 Hertz, and Figure 20 indicates that the horizontal response at the top of the superstructure is reduced greatly. However, if the service speed for the six-bladed propeller increases to 103 RPM, then blade frequency will equal the natural frequency of the superstructure. Reference 75 indicates the calculated vibration levels would be "very unpleasant" in this case.

23. Conduct Shaker Tests

Up to this point in the design process, all estimates of vibration levels have been based upon analytical response predictions. Since it is imperative the ship's actual response not exceed that established in the specifications, experimental tests should be conducted to excite the ship and measure the response before all finish work is completed. These tests hopefully will indicate any areas of structural deficiency, and corrective action can be taken before the ship's sea trials are begun. It should be

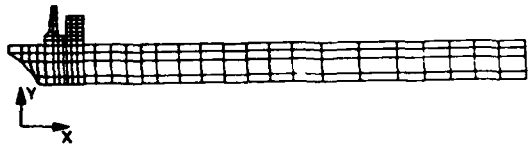


FIGURE 17. FINITE-ELEMENT MESH FOR TWO-DIMENSIONAL MODEL OF 370,000 DWT TANKER, FROM [75].

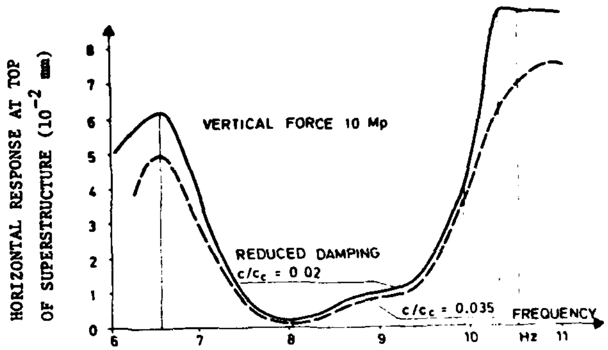


FIGURE 18. FORCED RESPONSE DEPENDING ON THE APPLIED GLOBAL DAMPING VALUE. 370,000 DWT TANKER, BALLAST CONDITION, FROM [75].

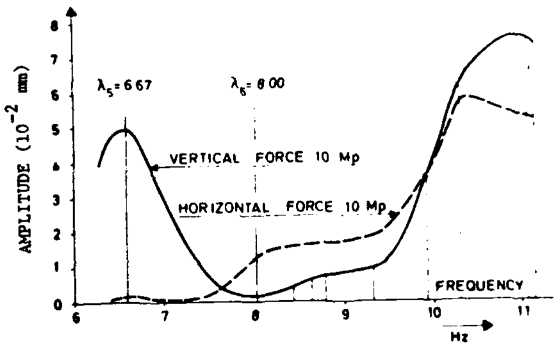


FIGURE 19. CALCULATED FORCED RESPONSE AT THE TOP OF SUPERSTRUCTURE IN LONGITUDINAL DIRECTION. 370,000 DWT TANKER, BALLAST CONDITION, FROM [75].

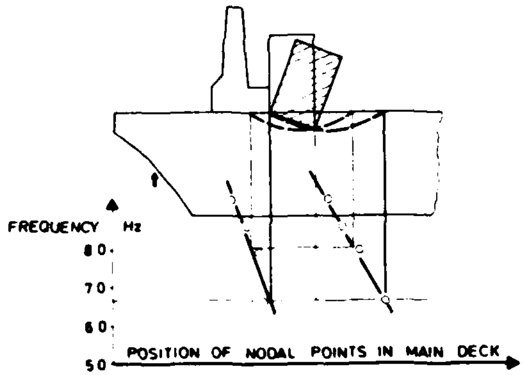


FIGURE 20. POSITION OF NODAL POINTS IN THE MAIN DECK FOR FORCED VIBRATIONS CALCULATIONS OF THE HULL GIRDER. 370,000 DWT TANKER, BALLAST CONDITION, FROM [75].

realized that vibration problems in the hull girder, engine room, or superstructure may be very difficult and expensive to correct at this point. The primary purpose of the shaker tests is to ensure the local plating and other local structures are properly designed. The shaker test can also be used to assess the accuracy of the structural modeling techniques and provide estimates of the ship's structural damping. These will be discussed in Design Block 24.

The tests should be conducted with the ship sitting in the water and the shaker mounted on a rigid foundation at the ship's stern. Surrounding disturbances should also be minimized so that the resulting vibration amplitudes can be related directly to the excitation forces. This may require conducting the tests at night when work on the ship is at a minimum and the water is calm. From experience in measuring vibrations generated in the bottom of a machinery space, it is believed that the stern shaker need not be large if the above conditions are met. The rigid foundation is necessary to excite global hull vibrations and not local vibrations in the vicinity of the shaker. The shaker should also be positioned as closely as possible over the propeller to more realistically simulate the propeller excitations. Figure 21, taken from Reference 94, shows the position of the exciter and some correlation of the experimental results with the finite-element model calculations. However, excitations should also be applied in the horizontal direction at the stern to assess the lateral-torsional behavior of the ship. Reference 94 describes the test procedures and results of shaker tests performed on a cargo-liner.

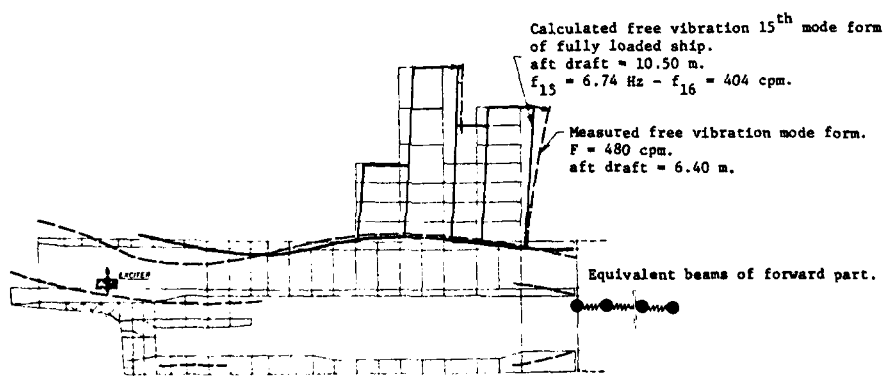


FIGURE 21. ELASTO DYNAMIC MODEL OF AFT PART AND CORRELATION OF EXCITER TESTS WITH FREE VIBRATION CALCULATIONS, FROM [94]

24. Assess Local Vibrations, Structural Damping, and Modeling Techniques

The shaker tests discussed in Design Block 23 provide a means of collecting some very valuable information about the vibration characteristics of the ship. By applying excitations at the stern and measuring the amplitudes and phase angles of the response, these tests will show if local resonances exist in the structure at the blade-rate frequencies or their

harmonics. The measurements should be taken at the points defined in the ship's specifications, as well as other places where local plating vibration could be present. If deficiencies do exist and can be corrected by local stiffening of the structure, these modifications are much easier and less expensive to make now rather than waiting until all finish work is completed.

The shaker tests also provide information on the amount of damping present in the structure. These estimates are usually obtained by computing the width of the resonant peaks. Reference 92 discusses this method along with the technique of impulsively exciting a ship and observing the decay of the free vibration modes. Anchor drops or underwater explosions are two methods of applying these impulsive loads. Although detailed knowledge of the damping will probably not help the design of the present ship, it will be valuable in future designs. These tests will help assess the influence that such variables as choice of materials, method of construction (e.g., welded or riveted), and type of framing have on the ship's damping characteristics. However, the tests will not provide any information on the damping due to the cargo because they are conducted in an unloaded condition.

The third way in which the shaker tests can aid the design process is to help estimate the accuracy of the structural modeling techniques. These tests will provide the measured response at various locations throughout the ship as a function of a known excitation at the stern. By applying excitations to the mathematical model of the complete ship developed in Design Block 21, the response can be calculated and compared with the measured values. This should provide a reliable means of judging the accuracy of the structural model since the input forces are accurately defined. The model can also use the damping computed earlier from the shaker tests.

Table 18 summarizes the information in this design step.

25. Measure Vibrations During Sea Trials

After the construction of the ship has been completed, the next step is to conduct the sea trials. The purpose of these trials is to determine how well the ship meets its performance specifications and identify any deficiencies in the design. This includes taking vibrations measurements at specified locations throughout the ship under known conditions so that these levels can be compared with the specifications defined in Design Block 1.

This report is not intended to serve as a guide for these vibration tests. References 95 and 96 are specifically written for this purpose. Reference 95 discusses vibration survey techniques, and an acquisition and processing system for ship vibration which can simultaneously measure about 60 different locations is described in Reference 96. Vibrations measurements to correlate with analyses should be much more detailed than those required by most codes such as given in References 6 and 7. As a word of caution, the vibration tests must follow a detailed test plan that ensures that all required vibration levels be measured. As a minimum, the plan should include:

TABLE 18. SUMMARY OF DESIGN BLOCK 24--ASSESS LOCATION VIBRATIONS, STRUCTURAL DAMPING, AND MODELING TECHNIQUES

Purpose:	To Determine if Local Resonances Exist in the Ship at Blade Rate Frequencies or Their Harmonics; To Provide a Means of Collecting Data on Hull Damping for Future Designs; To Estimate the Accuracy of the Modeling Techniques
Input:	Results of Shaker Tests Structural Model of Entire Ship
Output:	Location of Local Resonances in Structure Estimates of Structural Damping Quantitative Assessment of Accuracy of the Ship's Structural Model
References:	91 - 94

(1) Test Conditions

(2) Instrumentation Required

(3) Measurement Techniques

(4) Measurement Positions

(5) Data Reduction and Interpretation

(6) Report Requirements

Since there is only a limited amount of time allotted to vibration measurements during the sea trials, any confusion as to the test procedures will result in an incomplete set of vibration measurements.

26. <u>Compare Measured Vibrations with Specifications</u>

The real test of how well the ship was designed to minimize propeller-induced vibrations comes in this design block. This block is not related to design per se, but is part of the overall recommended design procedure. Its purpose is to assess quantitatively the acceptability of the ship's design with respect to propeller-induced vibrations.

All of the inputs necessary for making this comparison will be known at this point. The vibration specifications were defined in Design Block 1, and the corresponding vibration levels were measured during sea trials in Design Block 25. If the measured values are below those specified, then the ship's design is acceptable from the vibration point of view. If the measured values exceed the specifications, then the design is not acceptable and corrective action must be made. Responsibility for this

action will be contained in the vibration specifications if they are written correctly.

27. Compare Measured Vibrations with Calculations

Design Block 26 discussed how the acceptability of the ship was judged by comparing the measured vibrations with the specifications. If the vibrations levels are acceptable, then it could be assumed the recommended design procedures were adequate and were applied successfully. This may not be the case. There are instances of ships which should have been vibration free and had severe vibration problems. There are also examples where calculations predicted unacceptable vibration levels, but none were present when the ship was built. These point out that rather than the recommended procedures producing an acceptable design, the designers were simply lucky that the ship had no vibration problems. If the procedures are continued, then it is only a matter of time before a ship which vibrates badly will be produced. The last step in the design process is, therefore, to assess the process itself. The purpose of the assessment is to judge the accuracy and acceptability of the established design procedures and to identify areas in the design process in which improvements can be made. This can best be done by comparing the measured vibration levels taken during the sea trials with those computed for the entire ship assembly. A comparison of this type will provide the design team a quantitative description of how successful it has been in producing a ship with minimum propeller-induced vibrations. Let us assume, for example, the measured mode shapes of the hull girder compare favorably with those predicted, but the vibration amplitudes do not agree. This would indicate that the hull modeling procedures are correct, but the methods of calculating the propeller-generated forces and pressures are inaccurate and that improvements are needed in that area. This evaluation and feedback process will be discussed in the next chapter.

The writers of this report would like to emphasize again a previous point: the comparison between measured and calculated vibration levels should be made even though the ship had acceptable levels. This is usually difficult because of financial and time constraints, but is a necessary step in the design process if the recommended procedures are to improve continually. As a final comment, the results of this comparison should be disseminated to the maritime industry in some form. Again, this is difficult to accomplish, especially for ships which vibrate badly, because no organization wants to advertise it had part in such a design.

IV. DESIGN EVALUATION MILESTONES

In any design process, it is necessary to have checkpoints to determine if the design up to that stage is acceptable and should proceed or if modifications must be made. For simple problems, this procedure is informal and is continuously being done in the mind of the responsible engineer. However, in more complicated systems, such as a ship, where many individuals from different organizations and technical backgrounds are involved, the process needs to become more formalized. This is the purpose of the evaluation milestones shown in Figure 3. There are five such points during the ship's design procedure presented in this report, and they are designated as follows:

MILESTONE I - Preliminary Hydrodynamic Evaluation

MILESTONE II - Final Hydrodynamic Evaluation

MILESTONE III - Ship Substructure Evaluation

MILESTONE IV - Complete Ship Structure Evaluation

MILESTONE V - Test and Evaluation Review

Although the evaluation milestones are shown approximately at the end of each design phase, they are continuously being conducted throughout the particular phase. Only at the end of a phase would a formal report of the results be issued.

Since the evaluation steps are associated with a particular design phase, they serve to divide the design process into smaller tasks according to technical disciplines and make identification of problem areas easier. They also serve to establish milestone points for reporting the results and charting the progress of the entire project.

The five evaluation milestones discussed in the following sections are first presented by showing the overall flow diagram for each of the individual design phases. This diagram is then followed by a more detailed presentation of the various phases showing the evaluation and feedback processes. It is hoped that by utilizing this evaluation procedure, deficiencies in the ship's design which could cause excessive vibrations would be identified and corrected before the ship is built.

1. MILESTONE I - Preliminary Hydrodynamic Evaluation

Figure 22 shows the three blocks composing the preliminary hydrodynamic design. Block 2, in which the general ship design data are established, is not part of this phase, but is shown for completeness. An expanded description of the phase showing the feedback loops in the design process is shown in Figure 23. The reader should remember that the three primary objectives of this portion of the analysis are:

(a) Choose the number of propeller blades so that the blade-rate frequency is not near the longitudinal propulsion natural frequency.

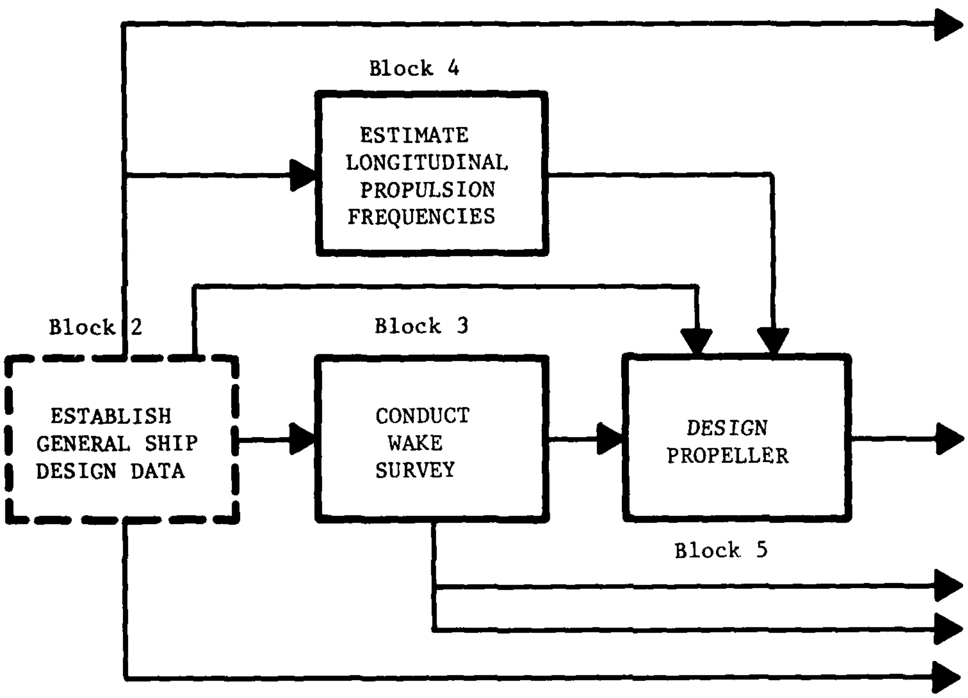


FIGURE 22. PRELIMINARY HYDRODYNAMIC DESIGN PHASE

(b) Choose a stern configuration whose wake distribution does not introduce large propeller and hull forces.

(c) Design a propeller whose efficiency is as high as possible.

These three objectives are reflected in decision blocks shown in Figure 23.

A very important part of this phase is to choose a suitable stern configuration because it determines the nature of the flow into the propeller. Since the propeller forces are dependent on the nonhomogeneous flow into the propeller disc, their magnitudes and directions are directly related to the wake distribution and hence the stern configuration. One way of graphically representing the wake is in terms of the Taylor wake number, W, defined as

$$W = 1 - \frac{V_a}{V_m}$$

where

V_a = axial velocity of water to propeller disc

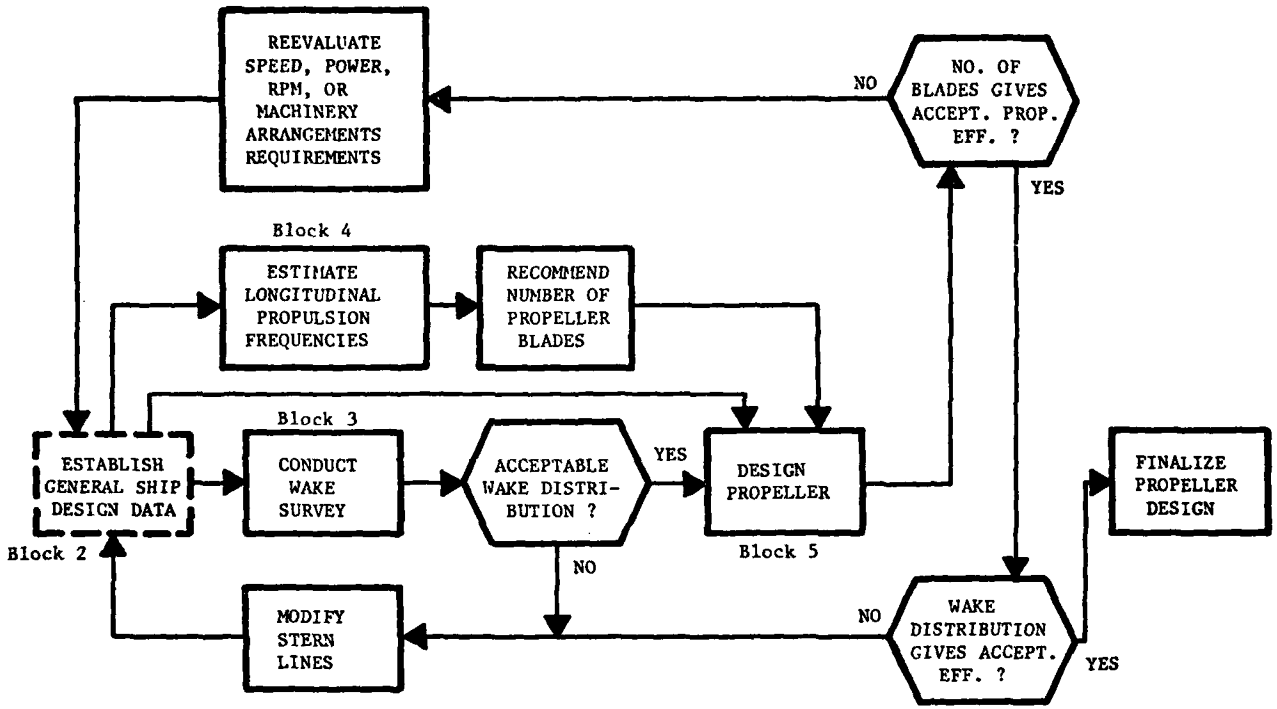
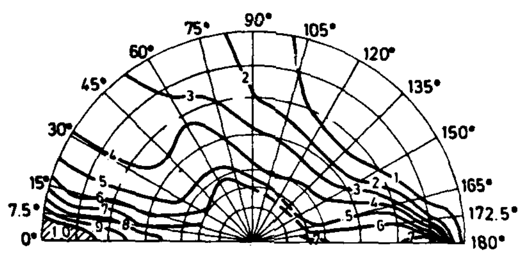


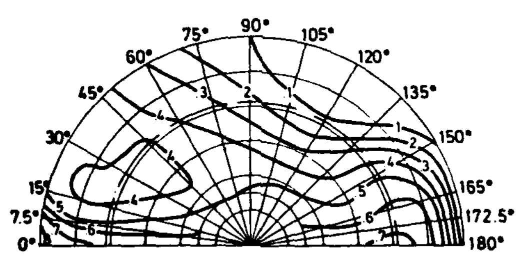
FIGURE 23. MILESTONE I - PRELIMINARY HYDRODYNAMIC DESIGN EVALUATION

V_m = velocity of model

The wake number is usually plotted on a polar map showing lines of constant W as shown in Figure 24 [97] for a particular stern configuration. This figure was taken from the axial wake distributions obtained in model tests for two aft body configurations. The zero degree angle is located at the top of the propeller. Figure 24(a) indicates that the inflow velocity is essentially not moving relative to the ship at the top position, and there is a steep wake gradient. From experience, this variation in wake would probably result in excessive excitations, and these excitations can be reduced by altering the ship's stern lines in the design stage. Figure 24(b) shows the wake distribution with the aft body modified, and the wake peaks and gradients are greatly reduced. Other examples of how propeller forces can be reduced by proper selection of the stern form and propeller type are given in a recent paper by Okamoto [98]. Solumsmoen [97] points out that it is difficult to give criteria for an acceptable wake because the excitations also depend on the propeller geometry. Huse [99] does suggest some guidelines, for example,



(a) Example of axial wake distribution for an original hull giving rise to great hydrodynamic forces



(b) Axial wake distribution for modified body lines

FIGURE 24. AXIAL WAKE DISTRIBUTIONS FOR ORIGINAL AND MODIFIED BODY LINES, FROM [97]

- For large tankers and other ships with high block coefficients, W_{max} should not exceed 0.75.
- For ships with small block coefficients, e.g., below 0.6, W_{max} should be below 0.55.

W_{max} is the wake number measured at the center plane of the propeller in the range 0.4 to 1.15 of the propeller's radius.

However, more research is needed in this area so that a systematic correlation between wake distributions and hydrodynamic excitations and propeller cavitation can be established. This correlation is dependent on the propeller geometry, and an acceptable wake does not guarantee low propeller-generated forces and no cavitation. But such information would provide the designer with a rational means of assessing the suitability of proposed aft body configurations.

2. MILESTONE II - Final Hydrodynamic Evaluation

After the design parameters in the preliminary hydrodynamic phase have been established and deemed acceptable in its evaluation, the final hydrodynamic design may be initiated. This phase was discussed in Sections 6-13 of Chapter III and is presented in Figure 25. The overall objective of the final hydrodynamic design is to be able to calculate all excitation forces produced by the propeller and to evaluate if these forces are likely to cause unacceptable levels of vibration in the ship.

The process by which the final hydrodynamic design is evaluated is illustrated in Figure 26. This figure includes the Design Step 6-13 along with the decision points and the feedback paths if redesign is deemed necessary.

The process begins by evaluating if the propeller forces computed in Block 6 are within acceptable limits. As discussed in Section 6 of Chapter III, there are no accepted guidelines for making this judgment, and additional research is required in this area to aid the designer. References 31 and 100 do provide some simple criteria for maximum permissible excitation, although they should be used with caution. If the magnitudes of the propeller forces and moments are not acceptable, then changes in the design must be made. These involve modifying the stern configuration to alter the wake distribution and/or changing the propeller design. If the stern lines are altered, then a new wake survey must be conducted to determine the new nominal wake field. This new wake field would in turn require that another propeller design be made. If the original wake is judged suitable, then perhaps only changes in the propeller's geometry are necessary. It is certainly desirable to redesign the propeller without altering the number of blades. However, if this is necessary, then a new longitudinal analysis of the shafting (Block 4) must be conducted.

After the propeller forces are judged acceptable, then the calculations for hull pressures, including the effects of cavitation, can be made. Chapter III indicated this could be done in either one of two ways. The indirect method consists of computing the hull pressures without cavitation,

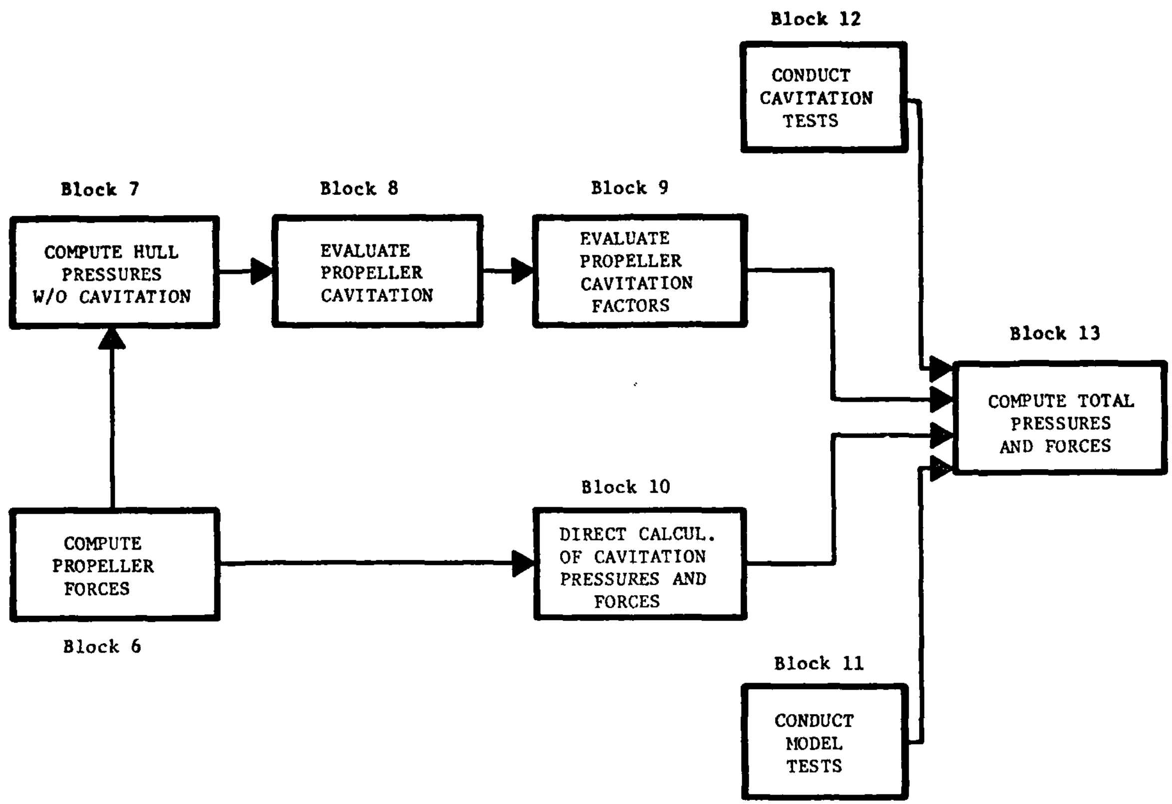


FIGURE 25. FINAL HYDRODYNAMIC DESIGN PHASE

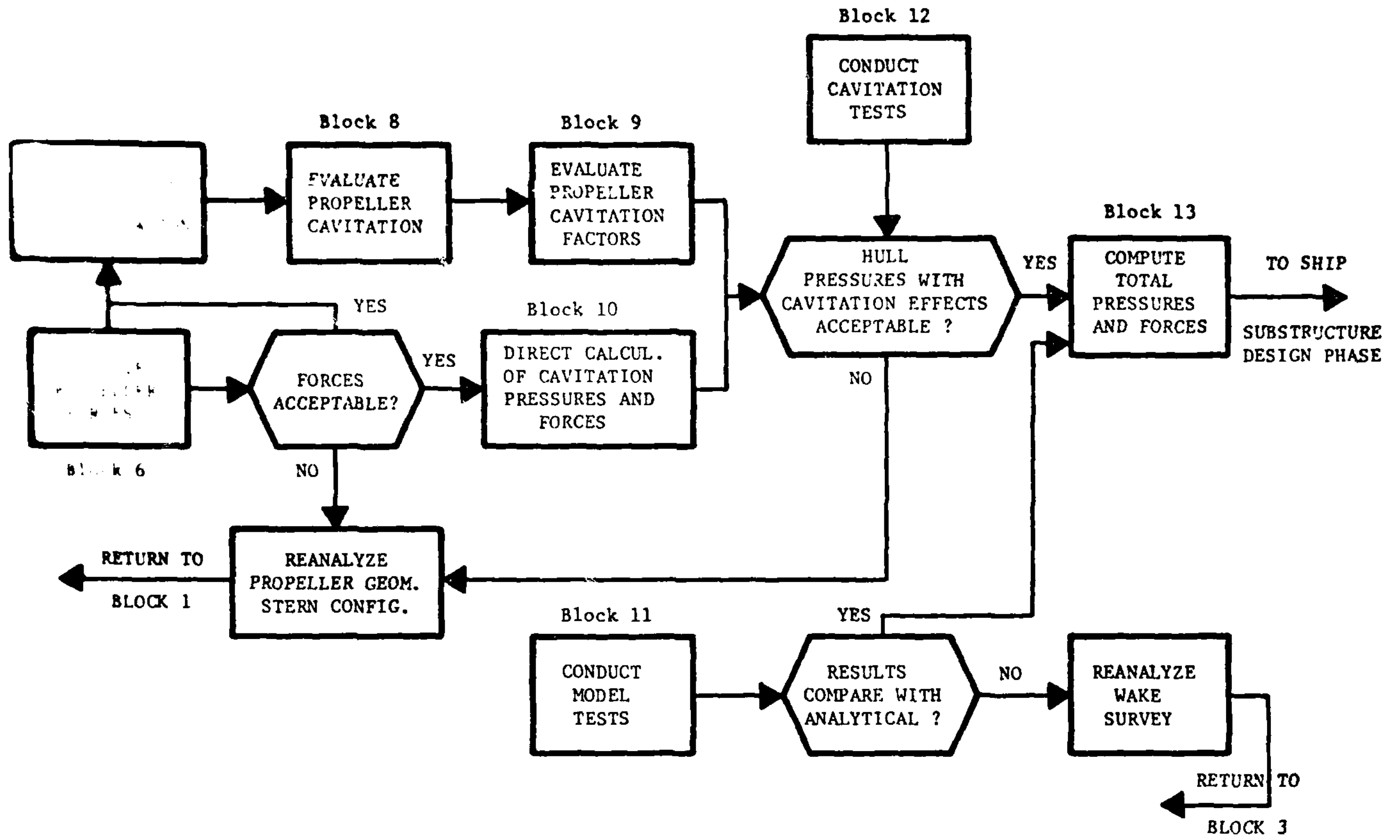


FIGURE 26. MILESTONE II - FINAL HYDRODYNAMIC EVALUATION

evaluating the effects of cavitation, and estimating the cavitation factors. This procedure was described in Blocks 7-9. The second method, Block 10, involves direct calculation of the cavitation pressures. It is certainly desirable to develop fully this direct process, and continuing research is being conducted in this area. The acceptability of the calculated hull pressures can now be judged. But as with the propeller forces, there is no accepted criterion. Huse [101] in 1971 gave the value of acceptable blade frequency pressure fluctuations p measured at a point directly above the propeller as

$$2p = 13{,}000 \text{ (Pascals)}$$

This figure was based on measurements from 12 ships. Some had vibration problems, and some did not [102]. Reference 102 reports that the Swedish State Shipbuilding Tank [103,104] has the following criterion for the pressure fluctuations above the propeller:

$$2P_r \leq 6250 \left(\frac{a_x}{a_z}\right)\left(\frac{\nabla}{10^3 D^2}\right) \text{(Pascals)}$$

where

$2P_r$ = peak value of pressure fluctuations at reference point

∇ = displacement, full load, m^3

D = propeller diameter, m

a_x = longitudinal clearance of propeller-hull, forward

a_z = vertical clearance of propeller-hull

The criterion was based on results of measurements made on eleven ships of different types. Reference 102 reports that the formula sometimes gives too low values of permissible excitations on small ships and proposes the following limit on single-screw merchant ships:

$$2P_{allowed} < \left(0.75 + \frac{75}{L}\right) P_r \quad \text{(Pascals)}$$

where

L = length of ship in meters

P_r = pressure amplitude calculated from previous formula

For detailed discussions of these criteria, the reader should see References 102-104.

The cavitation tests (Block 12) can also be conducted during this time either in a cavitation tunnel or in a depressurized towing tank. The

purpose of these tests is to study the extent of cavitation on the propeller blades and to measure the magnitudes of the pressure fluctuations on the hull. If the measured or calculated hull pressures are not acceptable by whatever criterion is chosen, then design changes must be made. Figure 26 shows that this again requires either modifying the stern configuration or propeller geometry.

The model tests (Block 11) are also conducted during the final hydrodynamic design. The purpose of these tests is to confirm the validity of the earlier wake survey by comparing measured and calculated propeller forces and moments. If agreement is obtained, then more confidence can be placed in the analytical hydrodynamic calculations of propeller-generated excitations because all predictions required a wake survey as input. If agreement is not obtained, then the original wake survey must be reanalyzed. This process is also shown in Figure 26.

3. MILESTONE III - Ship Substructure Evaluation

The purpose of the ship substructure design phase is to allow for the design, evaluation, and modification (if necessary) of each major subsystem before the complete structure is assembled and analyzed. This process allows the ship to be divided into smaller components for which the analysis will be simpler than that for the ship as a whole. If each substructure is properly designed, it is hoped the response will be acceptable when the complete ship model is assembled. Figure 27 shows a flow diagram of the ship substructure design phase, which was discussed in Sections 14-20 of Chapter III. This diagram is expanded in Figure 28 to include the various evaluation and feedback loops. These will now be discussed.

The first requirement is to determine if the resonant frequencies and forced response of the longitudinal shafting as determined by Design Block 14 are acceptable. As a guide, the resonant frequencies should not be within 25 percent of the blade rate frequency or its harmonics. Reference 5 and the design specifications can be used as a guide to permissible vibration levels. If the structure is deemed acceptable, then the stiffness and mass of the machinery space can be added to the model. If not, modifications must be made to the thrust bearing and foundation stiffnesses. The process is then continued with the machinery space included as shown in Figure 28. The same acceptability criteria can be used, with any modifications being made in the stiffness of the thrust bearing and machinery space.

The lateral response of the shafting with both rigid and flexible hull models is determined in Design Blocks 16 and 17. If the natural frequencies are again within about 25 percent of the excitation frequencies or if the bearing reactions are negative, it will be necessary to change the location of the bearings or alter the stiffness of the lateral supports at the ship's stern. This feedback process is illustrated in Figure 28.

The design of the superstructure is the next subsystem to be analyzed. Since the superstructure is excited only through the ship's hull girder, its evaluation is based on having the natural frequencies removed from the excitation frequencies. If this criterion is not satisfied, then the superstructure's natural frequencies must be changed, possibly by improving its continuity with the main deck structure. The philosophy of separating

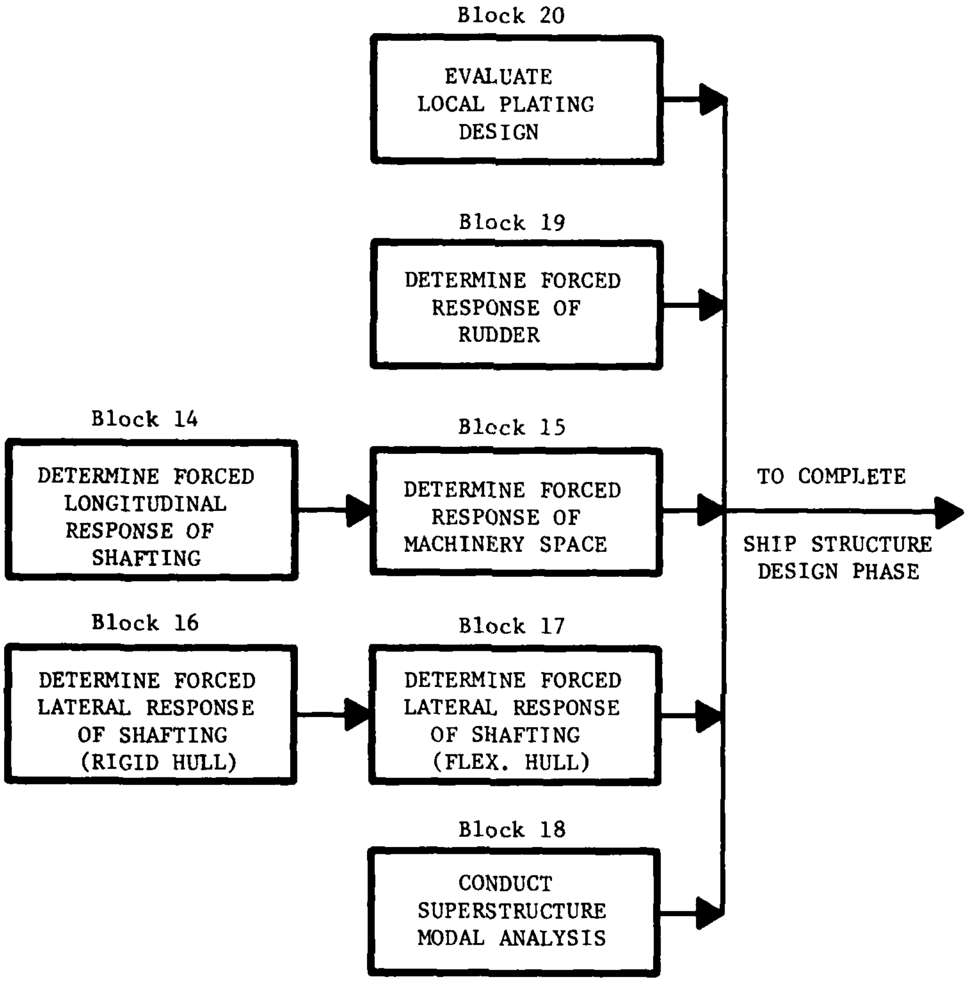


FIGURE 27. SHIP SUBSTRUCTURE DESIGN PHASE

the natural and excitation frequencies also applies in the design of the local plating. Modifications to the plating which did not meet the criterion would be made by local external stiffening and/or changes in plate thicknesses.

The final item in the ship substructure evaluation is the analysis of the ship's rudder. Again the primary means of judging the rudder's acceptability is based upon how close its natural frequencies lie to the propeller excitation frequencies. Figure 28 also shows that a criterion based upon forced response of the rudder should also be included. However, as discussed in Chapter III, the techniques for predicting propeller

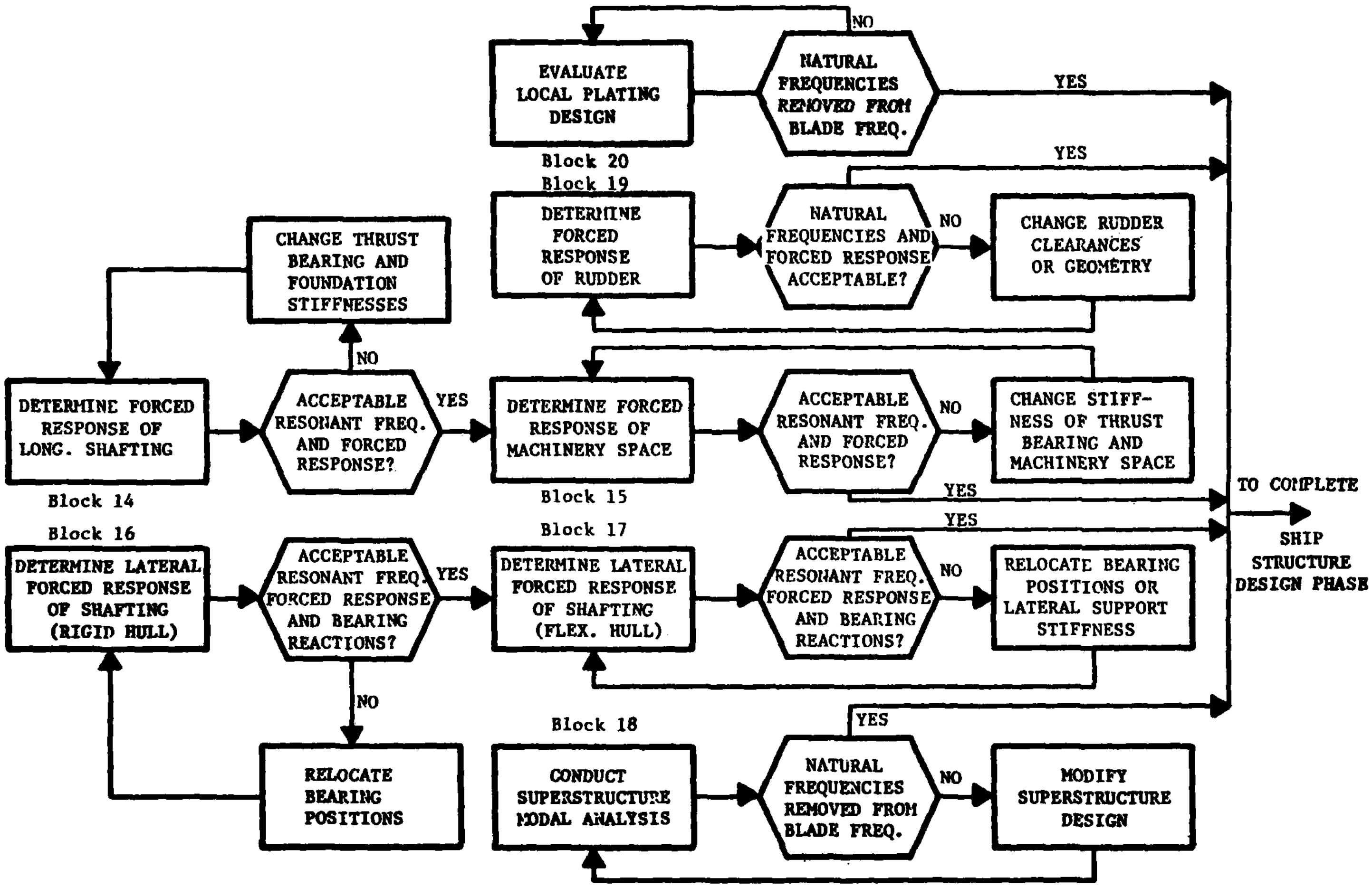


FIGURE 28. MILESTONE III - SHIP SUBSTRUCTURE EVALUATION

excitations are not well developed at the present time. Modifications to the rudder structure and its response must be made by changing its manner of support, geometry, or clearances with the ship's hull.

4. MILESTONE IV - Complete Ship Structure Evaluation

After the various subsystems analyzed in the ship substructure phase have been judged as acceptable, the next task is to conduct the complete ship analysis. This process is fairly straightforward and is depicted in Figure 29. The mathematical model of the ship is assembled from its components, loaded with the propeller-generated forces and pressures, and the response determined. The evaluation made in Milestone IV is based upon the magnitudes of the vibration amplitudes and stress levels compared with those set in the specifications. If the computed quantities are below the specifications, then the complete ship design is acceptable, and the design process may continue. If the specifications are exceeded, then the design is not acceptable and modifications must be made.

Figure 30 shows the complete ship structure evaluation process. In general, the vibration amplitudes are evaluated in each of the ship's major subsystems such as the hull girder, superstructure, machinery space, and propulsion shafting. It is again important to remember that the vibration and stress levels must be computed at the locations designated in the specifications. If deficiencies are found in the complete ship structure, then Figure 30 shows that design changes must be made in the deficient structures and a new analysis conducted. It is difficult to give any specific recommendations as to the possible structural modification which will correct a particular problem. If resonance occurs with the hull girder, it will be almost impossible to make structural modifications to avoid this resonance for all variations in ship mass and ranges of propeller RPM. Possible structural changes in the superstructure, shafting, and machinery space subsystems could, from past experience, include:

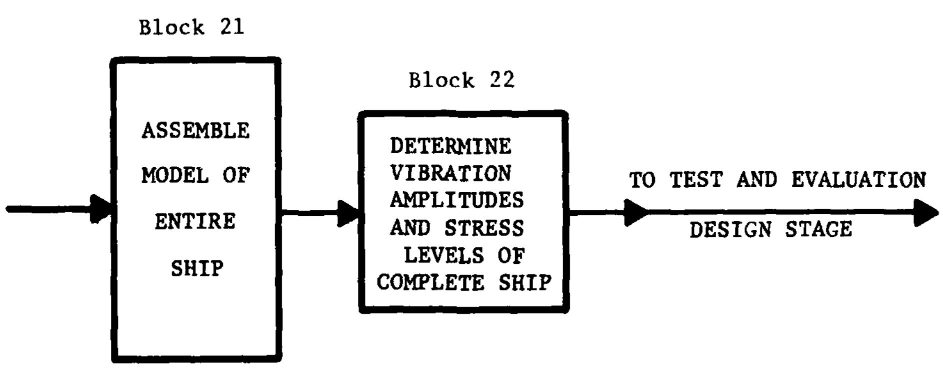


FIGURE 29. COMPLETE SHIP STRUCTURE DESIGN PHASE

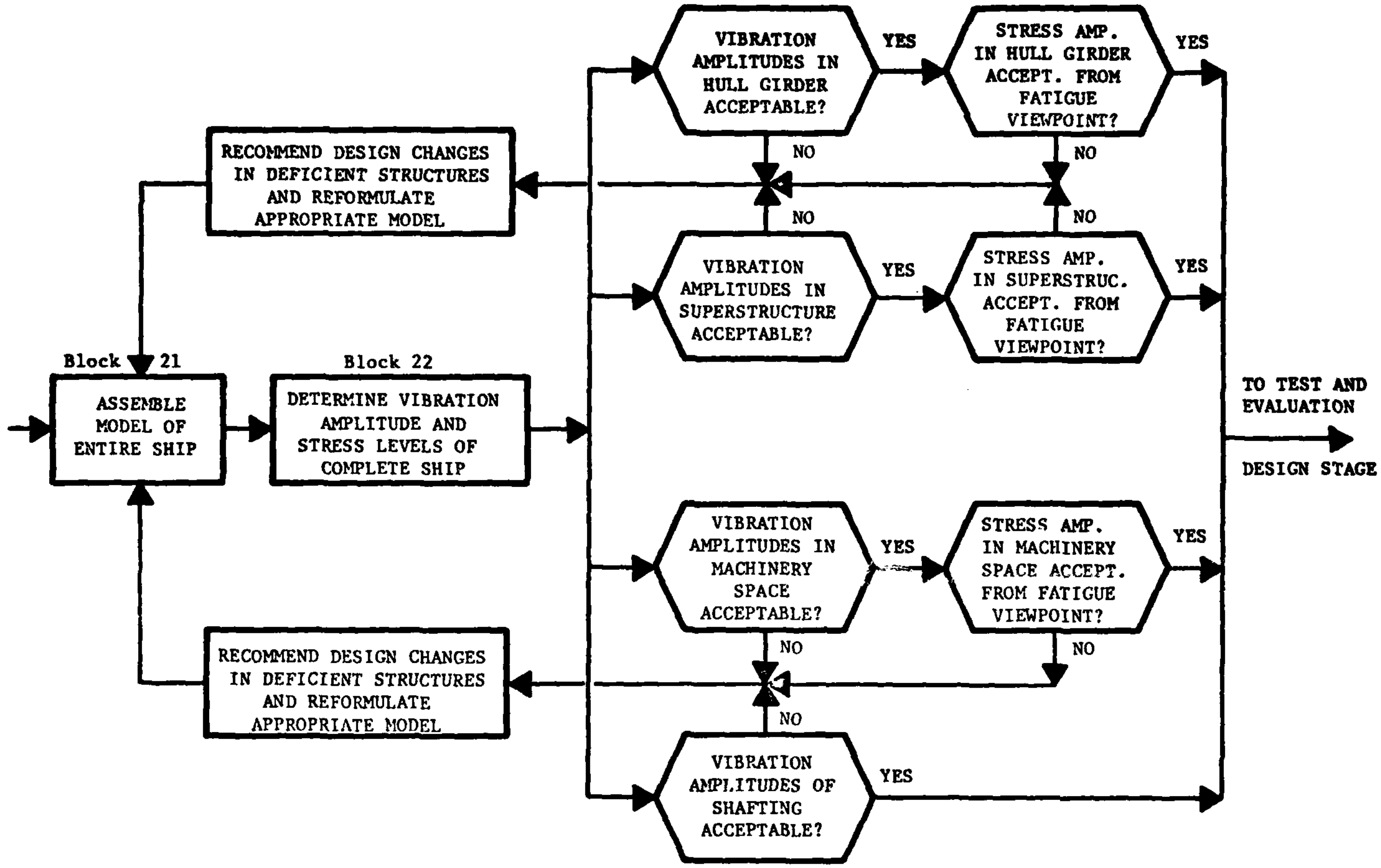


FIGURE 30. MILESTONE IV - COMPLETE SHIP STRUCTURE EVALUATION

Superstructure

. Plate Thicknesses
. Bulkhead Locations
. Structural Continuity with Main Deck
. Addition of Pillars
. Strengthen Beams
. Addition of Bulkheads

Machinery Space

. Bulkhead Locations
. Plate Thicknesses
. Foundation Stiffness
. Addition of Pillars
. Strengthen Beams
. Addition of Bulkheads

Shafting

. Lateral Support Locations
. Foundation Stiffnesses
. Diameter of Shafting

This portion of the feedback process must rely heavily on the analysts' and designers' correct assessment of the problem and possible corrective measures. Figure 30 is intended to show that structural changes should first be made in the deficient subsystem, with the hope of not significantly altering the response of the rest of the ship. This may not be possible since all systems are coupled together. In any case, structural changes must be incorporated into the complete ship model and the analysis reconducted. This process must continue until the structural response does not exceed the limits set in the specifications.

5. MILESTONE V - Test and Evaluation Review

The final phase of the recommended design procedure is for test and evaluation of both the ship and the design process. Figure 31 shows this phase, which was discussed in Sections 23-27 of Chapter III. Based on the input from the shaker tests, the local vibration levels, hull damping, and structural modeling techniques are determined. Of these three, the assessment of local vibration is the most important because it directly affects the acceptability of the current design. Hull damping estimates and the evaluation of the modeling techniques are also valuable, but more from the viewpoint of future ship designs and design methods. This is illustrated in Figure 32, which presents the review process for the Test and Evaluation Phase. The figure shows that if the local vibration levels are not acceptable as determined by the shaker tests, then the local structure must be modified before the ship is finished and sea trials begun. These modifications would usually consist of stiffening the deficient structure by such means as increased plate thicknesses or adding local stiffeners. However, if the predicted response at selected locations does not agree with the measured response to shaker excitations at the stern, then the structural

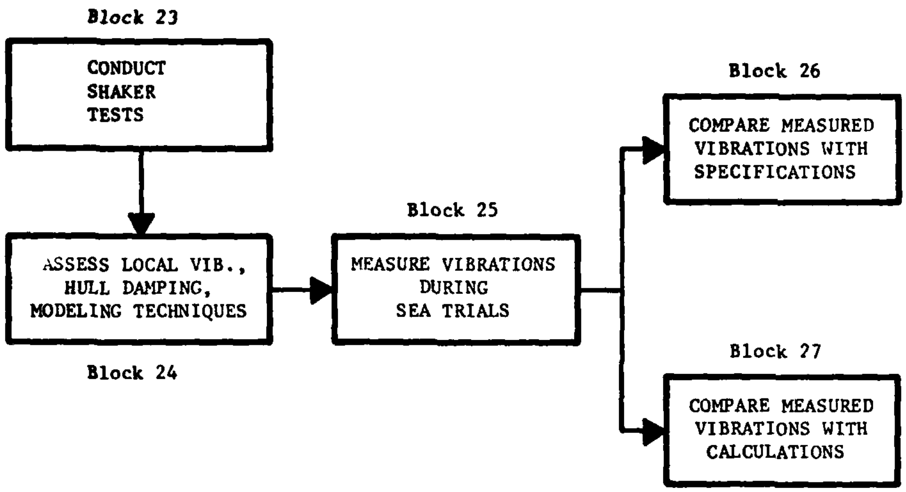


FIGURE 31. TEST AND EVALUATION DESIGN PHASE

model techniques are probably deficient and need to be improved. The design and construction process of the ship is not dependent on this review, and after the ship is completed, sea trials will begin.

In the sea trials, vibration levels will be measured at locations and under operating conditions given in the specifications. Figure 32 shows that the acceptability of the ship from a vibration point of view is based upon the measured levels meeting the specifications. If this is accomplished, then the design can be considered adequate. If the specifications are not met, then the design procedure has failed to produce an acceptable ship, and corrective measures must be taken.

This report discussed in Section 28 of Chapter III the fact that even though the vibration specifications were met, the design procedure may be inadequate. It is necessary, therefore, to end the process with an assessment of how well the complete design procedure works. This can best be done by comparing the calculated vibration levels with those measured during the sea trials. If sufficient agreement is obtained, then the recommended procedure can be considered adequate. If not, a review must be made to determine the source of errors. These errors arise essentially from two sources: (1) calculation of the hydrodynamic excitations and (2) calculation of the ship's response to these excitations. It is difficult to give any specific rules to pinpoint deficiencies in the analysis techniques. However, if good correlation is found between the calculated and measured response to the shaker excitations, the discrepancies are probably caused by inaccurate calculations of the propeller-generated forces and pressure rather than by poor structural modeling techniques.

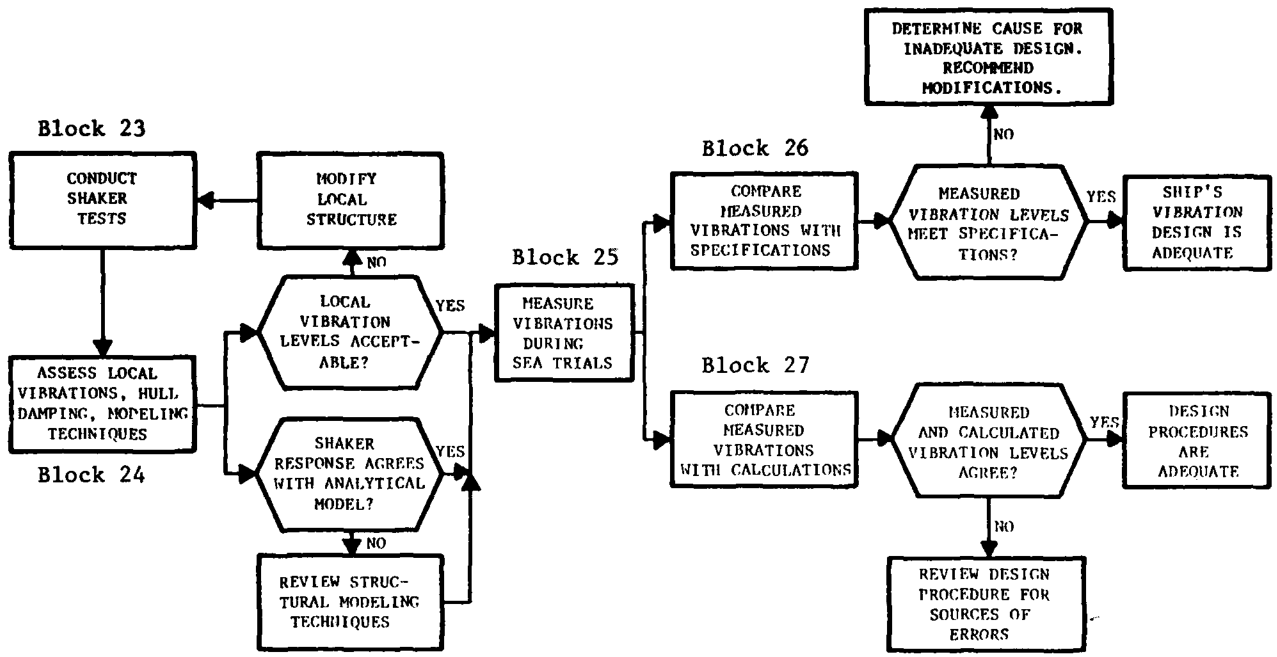


FIGURE 32. MILESTONE V - TEST AND EVALUATION REVIEW

V. APPLICATION OF THE RECOMMENDED PROCEDURE

1. Overview

It was recognized during the conduct of this program that all the detailed steps for vibration design recommended in Chapter III have not been applied to a single ship. Most designs use a portion of the individual blocks, but not necessarily in the sequence presented in this report.

In order to validate the recommended procedures to some degree, the original scope of work for this project was extended to include application to a specific ship. Detailed design information is difficult to obtain, but fortunately vibration analyses which follow closely the recommended procedures have been conducted by Littleton Research and Engineering Corp. (LR&EC), a corporate coauthor of this report. Section 2 presents the procedure as it is applied to a containerized and unitized cargo ship. This ship was built and underwent extensive vibration measurements. Appendix G contains a summary of vibration investigations which were part of the design of a proposed large roll-on/roll-off ship. This information is presented to show the depth of the studies which were made to minimize propeller-induced vibration. In both of these examples, the readers will immediately see that not all the blocks in the recommended procedures have been applied. Thus, only a partial example of application of the recommended design procedure could be given.

2. Procedure as Applied to a Containerized and Unitized Cargo Ship

2.1 General

To provide a comparison of the reliability of the recommended procedure, it was proposed to apply certain elements of the program approach to a ship built in 1972 by Ingalls Shipbuilding Division, Litton Systems, Inc., for Farrell Lines, Inc. The ship had a longitudinal vibration of the propeller and shafting system which required correction. Because of these difficulties, an unusually complete set of vibration readings were made on the ship. LR&EC had made quite extensive vibration studies of the ship in the design stage, but the studies lacked the completeness and perspective of the program presented in this report. The studies included a rough finite-element analysis of the machinery space, which indicated a double bottom mode. This mode when coupled with the shafting system represented the offending vibration. However, under the pressures of time and finances, this difficulty was not recognized at the time. The following sections update certain portions of the original analyses and compare the results with test measurements.

2.2 Ship Description

The important ship characteristics are listed below:

Overall length	668.5 ft
Length between perpendiculars	625.0 ft

Beam	90.0 ft
Depth to main deck	53.0 ft
Design draft	29.0 ft
Displacement (at 31' 5" draft)	28,440.0 tons
Number of containers	872

The ship is divided by transverse bulkheads into eight holds forward of the machinery space and one long hold aft. The quarters and bridge are located over the machinery space. There are wide hatches port and starboard of a centerline girder in which the containers are stowed vertically in guides. The containers are stowed six high in the holds and two high over the hatch covers.

Longitudinal strength and some torsional stiffness of the hull are provided by a heavy box girder 9 ft deep by 9 ft 9 in. wide extending between the second and main decks. In addition, there are three heavy H beams on each deck, and portions of the ship obtain torsional stiffness from wing tanks. Extensive use is made of high-strength steel.

The ship is driven by a steam turbine plant with a normal horsepower rating of 26,000 and an ABS rated horsepower of 28,500. The corresponding propeller shaft speeds are 103 RPM and 106.5 RPM, respectively. The overspeed trip on the turbines is 115 percent of the maximum rated speed, which corresponds to 123 RPM for the propeller. The normal operating speed of the ship is 23 knots. Reference 105 contains detailed information on this class of containership.

The propeller is 23 ft 6 in. in diameter and has six blades. It is made of Superston 40, ABS Grade 5, whose tensile strength is 90,000 psi. The propeller shaft is supported by oil-lubricated bearings. The aft bearing has a diameter of 29-3/8 in. and a length of 74 inches. The forward bearing is 29-1/2 in. in diameter and 25-1/2 in. long. The distance between the bearings is 24 ft 9 in., and the distance from the center of the aft bearing to the centerline of the propeller is 6 ft 9 inches.

The choice of clearances between the propeller tip and the hull is determined by a balance of several factors. From a structural point of view, it is desirable to support the propeller with little overhang and minimum extension from the hull. The hydrodynamicist would prefer to have the propeller work in the high wake boundary layer region close to the hull for good propulsive efficiency. For low levels of propeller-excited vibration, however, it is desirable to have large propeller clearances. The intensity of the pressure field around the propeller falls off rapidly with distance, and the propeller works in a more uniform wake. Consequently, smaller harmonic propeller and surface forces are generated.

In the preliminary design, the designer normally defines the propeller location in terms of clearances between the propeller and adjacent structure expressed as percentages of the propeller diameter. For this ship the clearance to the hull over the tip of the blade is about 3.21 ft or 13.6 percent of the propeller diameter. The distance from the propeller leading edge at 0.7 radius to the strut is about 6.85 ft or 29 percent of the propeller diameter. The distance from the propeller trailing edge at 0.7 radius to the rudder horn is 4.09 ft or 18 percent of the

propeller diameter. These are adequate clearances for the high power. The propeller is located 5 in. forward of Frame 223, or 616.58 ft from the forward perpendicular, or about 95 percent of the waterline length from the forward perpendicular.

An outboard profile of the ship is shown in Figure 33 and a sketch of the propeller aperture in Figure 34.

2.3. Application of the Procedures

The following paragraphs discuss the individual blocks of the Recommended Design Procedures which are covered in this application. The last series of digits refer to Figure 3, the Flow Diagram of Recommended Design Procedures to Minimize Propeller-Induced Vibrations, on pages 6 and 7 of this report.

2.3.1. Define Vibration Specifications. No quantitative specifications for acceptable level of vibrations were established in the ship construction contract. The contract specifications for the construction of the ship contained vibration clauses from the Maritime Administration Standard Specifications for Cargo Ship Construction. In connection with its analytical predictions, Littleton Research and Engineering Corp. developed some criteria. In analyzing the test results, the International Standards Organization ISO 2631, "Guide for the Evaluation of Human Exposure to Whole Body Vibration," was used. The reader is referred to Reference 4 for later revision of the standard.

2.3.2. Establish General Ship Design Data. The more significant characteristics of the ship were presented earlier. Sketches of the machinery space and the superstructure will be presented in the description of the model that is analyzed. The analysis is made for the ship at trial draft conditions in order to compare with measurements.

The drafts and displacements on trials were:

	Draft Forward	Draft Aft	Trim	Tons
Official Trials	19' 9"	26' 5"	6' 8"	19,442
Standardization Trials	20' 7"	26' 1/2"	5' 5-1/2"	19,261

2.3.3. Conduct Wake Survey. A wake survey was conducted by DTNSRDC in March 1968. The longitudinal, tangential, and axial wakes at 0.335 R (radius), 0.520 R, 0.723 R, 0.950 R, and 1.100 R are presented as Figures F1 - F5 in Appendix F. These wakes are for 25,035 tons displacement and 23 knots.

2.3.4. Estimate Longitudinal Propulsion Frequencies. Figure 35 plots the relationship between natural frequency of longitudinal vibration and the stiffness of the thrust bearing foundation. Kane and

FIGURE 33. OUTBOARD PROFILE

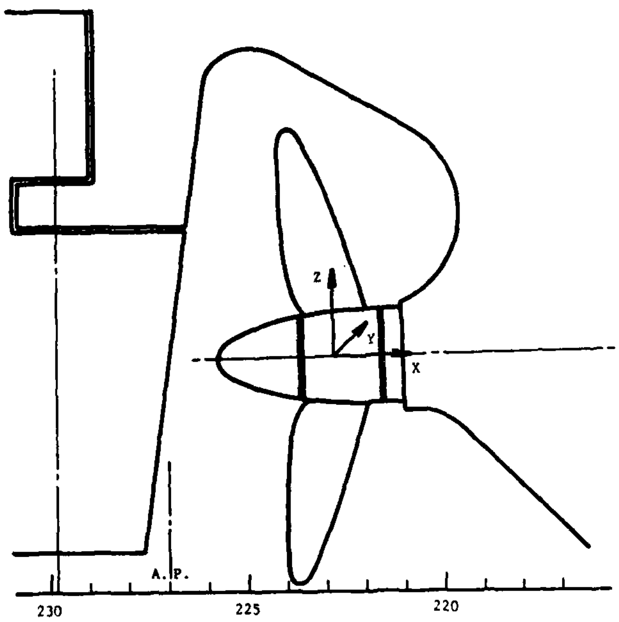


FIGURE 34. PROPELLER IN APERTURE

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A DESIGN PROCEDURE FOR MINIMIZING PROPELLER-INDUCED VIBRATION I--ETC(U)
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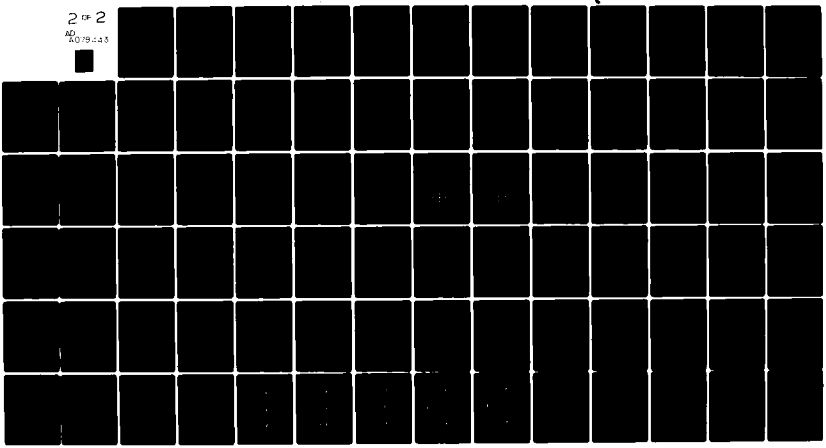

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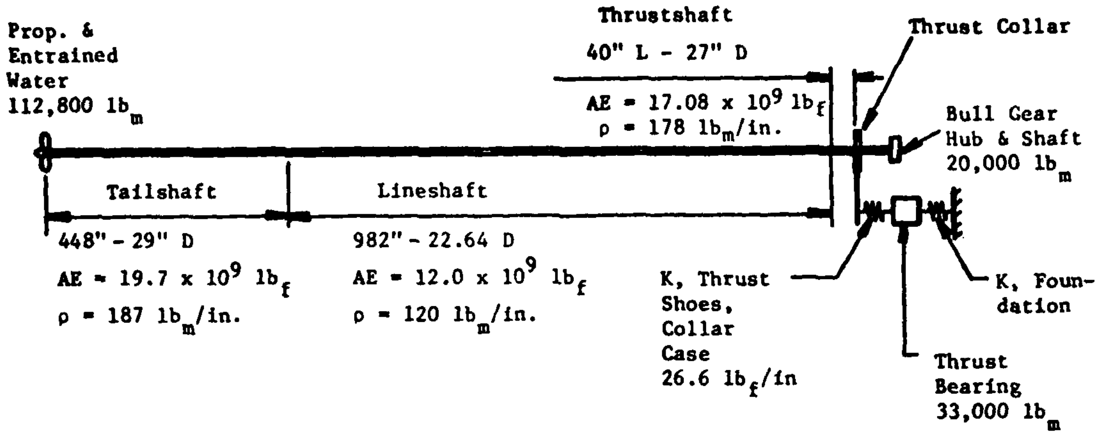
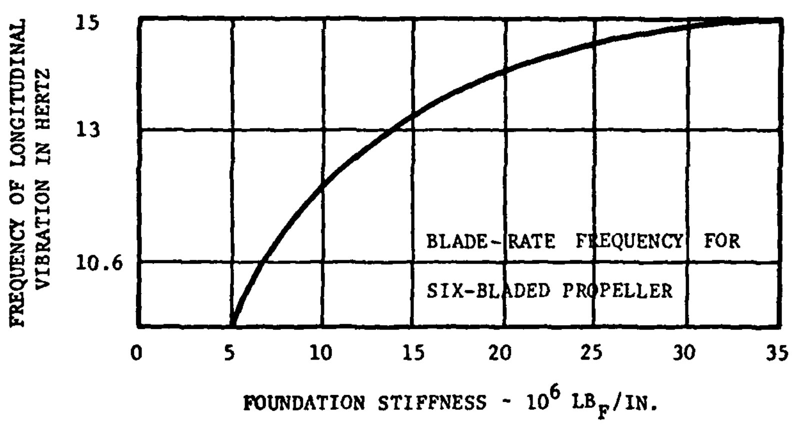


(a) Propeller and Shaft Mass-Elastic System



(b) Estimate of Natural Frequency for
Propeller Shafting System

FIGURE 35. FREQUENCY OF LONGITUDINAL VIBRATION VERSUS FOUNDATION STIFFNESS

McGoldrick [13] give some approximate ranges for K, the foundation stiffness, for a thrust bearing located just abaft the gear and tied into gear foundation, as 5 to 20 x 10^6 lb/in. On the basis of this, the designers were informed early in the design process that if a rigid foundation were designed (this should not be too difficult because the thrust bearing is located in a narrow region of the ship), there should be an adequate margin between the natural frequency of the shafting system and the operating frequency of a six-bladed propeller turning at 106.5 RPM. The foundation was well designed and of heavy construction, so by the general criteria applied at the time, longitudinal vibration trouble should not be expected. Why the trouble occurred will be explained later. The prediction of the trouble required additional analysis of the engine room.

2.3.5. Design Propeller. The propeller design was furnished to the ship builder along with the lines plan and general arrangement of the ship as part of the contract documents.

2.3.6. Compute Propeller Forces. The blade order harmonic forces and moments were computed using the LR&EC lifting line computer program. It was not possible to compute the twice-blades order forces and moments because only the first ten harmonics of the wake were furnished. Computations of twice-blade order forces and moments for a six-bladed propeller require the eleventh, twelfth, and thirteenth orders.

The values of the blade order harmonic forces and moments, with their phase angles, excited by the propeller on the shaft at 99 RPM are:

Fx (longitudinal)	12,963 lb	65.53°
Fy (transverse	4,210 lb	-19.70°
Fz (vertical)	1,185 lb	-128.51°
Mx (about rotational axis)	40,373 ft-lb	63.47°
My (about transverse axis)	99,363 ft-lb	-11.42°
Mz (about vertical axis)	37,210 ft-lb	-106.25°

All forces and moments are referenced to the coordinate system shown in Figure 34, and all moments are computed about the origin. The angles are measured from the top of the propeller, with the positive values in the direction of propeller rotation.

2.3.7. Compute Hull Pressures Without Cavitation. The hull pressures and the resulting hull force were computed using the LR&EC program. This program computes the free-field pressures generated by the distribution of forces along the lifting levers of the propeller blades and the thickness of the propeller blade and multiplies the result by a factor, generally 2, to represent the solid boundary.

Because at the trial conditions the draft aft was between 26 ft and 26.5 ft, the waterline runs out within the propeller aperture between the propeller and the rudder. Therefore, there is little hull surface below water in the region of the propeller. The predicted hull forces generated by the propeller at 99 RPM, if the propeller is not cavitating, are:

Fx	3,284 lb	-166°
Fy	2,430 lb	70°
Fz	4,310 lb	-180°
Mx	36,706 lb-ft	-111°
My	28,470 lb-ft	128°
Mz	19,112 lb-ft	73°

The following blocks are combined:

2.3.8. Evaluate Propeller Cavitation

2.3.9. Evaluate Propeller-Cavitation Factors

2.3.13. Compute Total Pressures and Forces

LR&EC has developed a computer program which predicts the pressures on the hull of a ship generated by the lift forces and the thickness of the propeller moving through the water. It does this by computing the pressure that would be generated in open water at the location relative to the propeller that represents a point on the hull and multiplying this free-field pressure by two to account for the rigid-hull surface. This is an approximation because it does not deal with the complexities of the reflections off the hull surface and does not include cavitation effects.

The pressure radiation from cavitation is much more efficient than that from the harmonic force loadings and from the passage of the blade through the fluid. In cavitation there is a harmonic change in the volume of the cavity. This transmits pressure as a simple source with the pressure amplitude at large distances from the source given by the equation (see Reference 106, page 313)

$$p = \frac{\rho}{4\pi r} \frac{d^2V}{dt^2}$$

where

$\frac{d^2V}{dt^2}$ = the second time derivative of the volume

ρ = the mass density of fluid

r = the radius from the source.

The harmonic forces on the blade and the passage of the blade through water do not change the volume, but only displace the water, and the pressure field corresponds to that from a source and sink, a dipole. The equation for the amplitude of the pressure for a harmonic force at a large distance from the dipole is (see Reference 106, page 318)

$$p = \frac{3F\nu \cos\theta}{2cr}$$

where

p = pressure

F = force on the sphere in the direction of vibration

ν = frequency in cps

θ = angle between a vector to the measuring point and the force vector

c = the speed of sound in the fluid

r = distance between the dipole's center and location of desired pressure

It is clear that the pressure from a harmonic force, as a source and sink in general, is a maximum in the direction of the force and zero in a plane normal to the force direction. On the other hand, the pressure from the changing volume has no directional properties.

At the present time, the procedures for predicting the hull pressure from a changing cavitation bubble are not well developed and are complicated. For this analysis, the loads from pressures on the hull surface, required as input to the hull vibration calculation, are estimated by comparing measured pressures on two ships with propellers at times working in the cavitation region, with the calculated pressures without cavitation determined by the procedure described earlier. These two ships are an OBO ship of the San Clemente Class built by National Steel and Shipbuilding Corp. and a RO/RO ship of the Seabridge Class built by Ingalls Shipbuilding Corporation [107].

The OBO ship has the following characteristics:

Full Power - 24,00 HP
Ship Speed - 17.15 knots
Propeller RPM - 92
Propeller Diameter - 26 ft
Propeller Projected Area - 265.04 ft^2
Propeller Tip Submergence - 18' 10-1/2"
Propeller Thrust - 420,000 lb
Mean Inflow Velocity Ratio at 0.8 R - 0.643
Inflow Velocity Ratio at 0.8 R, 0° - 0.306

Pressure readings were taken under steady ahead operation at 60 RPM and at 92 RPM. At 60 RPM the average ratio of 11 measured pressures to calculated pressures was 1.592 and the standard deviation of this ratio, 0.718. At 90 RPM the average ratio of 11 measured pressures to calculated pressures was 3.053 and the standard deviation of this ratio, 1.435.

Measured pressures and information on the harmonic forces are reported in Reference 107.

The Seabridge Class ship has the following characteristics [22]:

Full Power - 30,000 HP
Ship Speed - 24.59 knots
Propeller RPM - 112.9
Propeller Diameter - 23 ft
Propeller Projected Area - 268 ft^2
Propeller Tip Submergence - 7.9 ft
Propeller Thrust - 415,860 lb
Mean Inflow Velocity Ratio at 0.8 R - 0.866
Inflow Velocity Ratio at 0.8 R, 0° - 0.405

Readings of the hull-surface pressure were made under steady ahead operation at 110, 108, 101.2, 95.8, 92, 80, and 70 RPM [108]. At the higher speeds there was considerable hull vibration, so the pressure readings were corrected for water inertia effects by assuming the hull an infinitely long cylindrical body having a width corresponding to the waterline width in the plane of the pickup. The water correction is significant, but, on further reflection, the amount of the correction is considered to overcompensate, and half as much compensation would be more reasonable. The pressures measured at 70 RPM were so small as to be considered unreliable (maximum calculated pressure 74 lb/ft^2 = about 0.5 psi, minimum calculated pressure 4.3 lb/ft^2 = 0.03 psi). The remaining results are given in the following table.

TABLE 19. PRESSURE DATA AT 16 LOCATIONS ON SEABRIDGE SHIP

RPM	Accel. Correct.	Maximum Measured Pressures (lb/ft^2)	Maximum Calc. Pressures (lb/ft^2)	Max.* Ratio	Min.* Ratio	Avg. Ratio	Standard Deviation σ
110.0	No	432	183.7	9.44	0.92	3.073	2.139
110.0	Yes	585	183.7	14.77	1.43	5.813	3.473
108.0	No	432	177.1	10.39	1.02	3.447	2.250
108.0	Yes	637	177.1	22.14	1.71	5.688	4.160
101.2	No	245	154.9	7.69	0.82	2.313	1.646
101.2	Limited	255	154.9	7.69	1.07	2.445	1.601
95.8	No	208	138.8	5.31	0.78	1.965	1.184
92.0	No	196	128.0	4.53	0.41	1.682	1.057
80.0	No	99	96.8	22.08	0	0.941	--

*Maximum (minimum) ratio of measured to calculated pressures

The characteristics of the ship being analyzed are:

Design Shaft Horsepower - 26,000 HP
Ship Speed at 19,500 Ton (Model Test) - 25.53 knots
Corresponding Propeller RPM (Model Test) - 106.7
Propeller Diameter - 23.5 ft
Propeller Projected Area - 243.2 ft^2
Propeller Tip Submergence - 2.71 ft
Propeller Thrust - 354,860 lb
Mean Inflow Velocity Ratio at 0.8 R - 0.870
Inflow Velocity Ratio at 0.8 R, 0° - 0.393

To utilize the experimental data for the new ship, it is necessary to develop some method of comparison. It is presumed that, in general, the pressures integrated over the surface of the ship to give the total harmonic force are adequate for the noncavitating condition. These harmonic forces and moments can be multiplied by a factor to account for the cavitation. This is a simplification since, as shown earlier, the distribution of pressures due to cavitation is different from that due to blade forces and displacement.

This simplification is worse for pressures than for forces. In general, the forces due to loading the propeller act axially and tangentially on the propeller and would, therefore, tend to give small pressures where the plane of the propeller intersects the hull surface. This is the region of peak pressures because of proximity to the propeller. The cavitation effects being transmitted without any directional bias would give higher ratios of (pressures with cavitation)/(pressure without cavitation) in the regions close to the plane of the propeller disc. The hull forces, being an integration over the surface, average the variations.

It is further presumed that some of the pressure available before cavitation occurs will be used by the steady lift of the propeller. The remaining pressure will be available for pressure fluctuations, and if these fluctuations are greater than that remaining, cavitation will occur. Because the object is correlation of experimental data, hopefully the procedures need not be precise. For example, average pressures rather than peak pressures are considered. Because cavitation is a tip phenomenon, 0.8 R is taken as the propeller blade section for study. The submergence plus atmospheric pressure at the propeller's top on the centerline at 0.8 R is determined (neglecting thermal effects on vapor pressure). The thrust per square foot of projected area is obtained by dividing the thrust by the projected area of the propeller. The difference is available for handling fluctuations in wake. The pressure corresponding to the major fluctuation, the high wake at the top of the propeller disc, is determined very roughly by determining the change in angle of attack $\Delta\alpha$ at this location. The value is

$$\Delta\alpha = [\text{mean axial wake velocity} - \text{local wake velocity}] \times \frac{1}{\omega R_{0.8}}$$

$$= (V_{\text{mean ratio at } 0.8\,R} - V_{\text{ratio at } 0.8\,R,\,0^\circ}) \times V_{\text{ship}}$$

If the slope of the coefficient of lift curve at zero angle of attack is taken to be 2π, the average pressure on the section is

$$p = 2\pi\alpha \times \frac{1}{2}\rho V^2$$

$$= 2\pi\alpha \times \frac{1}{2}\rho\,[(\omega R_{0.8})^2 + V^2_{axial}]$$

We note that V^2_{axial} is negligible compared with $(\omega R_{0.8})^2$. Therefore, the change in pressure is

$$\Delta\text{Pressure} = [(V_{\text{mean ratio at } 0.8\,R} - V_{\text{ratio at } 0.8\,R,\,0°}) \times V_{ship}$$

$$\times 1.688] \times \frac{1}{2} \times \frac{64}{32.2} \times \frac{RPM}{60} \times 0.4\,D$$

$$= 0.01118 \times \Delta\text{Velocity ratio} \times V_{ship} \times RPM \times D$$

where V_{ship} is the ship's speed in knots and D is the propeller diameter in feet.

When these relations are applied to the full-power condition with the OBO and RO/RO ships, the values shown in Table 20 are obtained.

The pressure magnifying factors for the RO/RO and the OBO ship are plotted against the cavitation factor in Figure 36. There is clearly no agreement between the OBO curve and the RO/RO curve, and although several hypotheses were tried in an attempt to find a correlation, there appeared to be no way to obtain a common relation compatible with both.

For the present, the unitized and containerized ship being studied is similar to the Seabridge RO/RO ship, and so the pressure magnification as a function of cavitation factor for this ship is applied to the calculated hull forces and moments for the ship being analyzed. The vertical hull forces and transverse moments calculated at 99 RPM are

		Force	Phase Angle
Fz	=	4,310 lb	-180°
My	=	28,470 ft-lb	128°

These values are small because most of the hull is out of the water. By taking the forces and moments proportional to the square of the RPM, values at other RPM's can be calculated. These are given in Table 21, along with the forces and moments obtained by applying the cavitation factor. The values corrected for cavitation are plotted in Figure 37, and these values will be applied to the model developed and discussed in Sections 2.3.15 and 2.3.18.

TABLE 20. COMPUTED CAVITATION FACTORS FOR THE THREE SHIPS

RPM	Steady Pressure Thrust, lb/ft^2	Submergence Pressure, Minus Steady Pressure lb/ft^2	ΔPressure due to Wake, lb/ft^2	Cavitation Factor Col. 4/Col. 3
Seabridge Ship				
Submergence Pressure at 0.8 R = 2767 lb/ft^2				
112.9	1552	1215	329	0.273
110	1473	1294	312	0.241
108	1420	1347	301	0.223
101.2	1247	1520	264	0.174
95.8	1117	1640	237	0.145
92	1031	1736	219	0.126
80	779	1988	165	0.083
San Clemente OBO				
Submergence Pressure at 0.8 R = 3490 lb/ft^2				
92	1452	2038	155	0.076
60	618	2872	66	0.023
Containerized and Unitized Cargo Ship				
Submergence Pressure at 0.8 R = 2441 lb/ft^2				
106.7	1459	982	341	0.348
100	1282	1159	300	0.259
95	1157	1284	270	0.211
90	1038	1403	243	0.173
85	926	1515	217	0.143
80	820	1621	192	0.118

The following three blocks were not accomplished:

2.3.10. Direct Calculation of Cavitation Pressures and Forces

2.3.11. Conduct Model Tests

2.3.12. Conduct Cavitation Tests

2.3.14. Determine Forced Longitudinal Response of Shafting. Although normally this improvement upon the estimate made earlier (Block 4) should be made in the design procedure, this calculation will be included with the response of the machinery space and superstructure (Blocks 14 and 18).

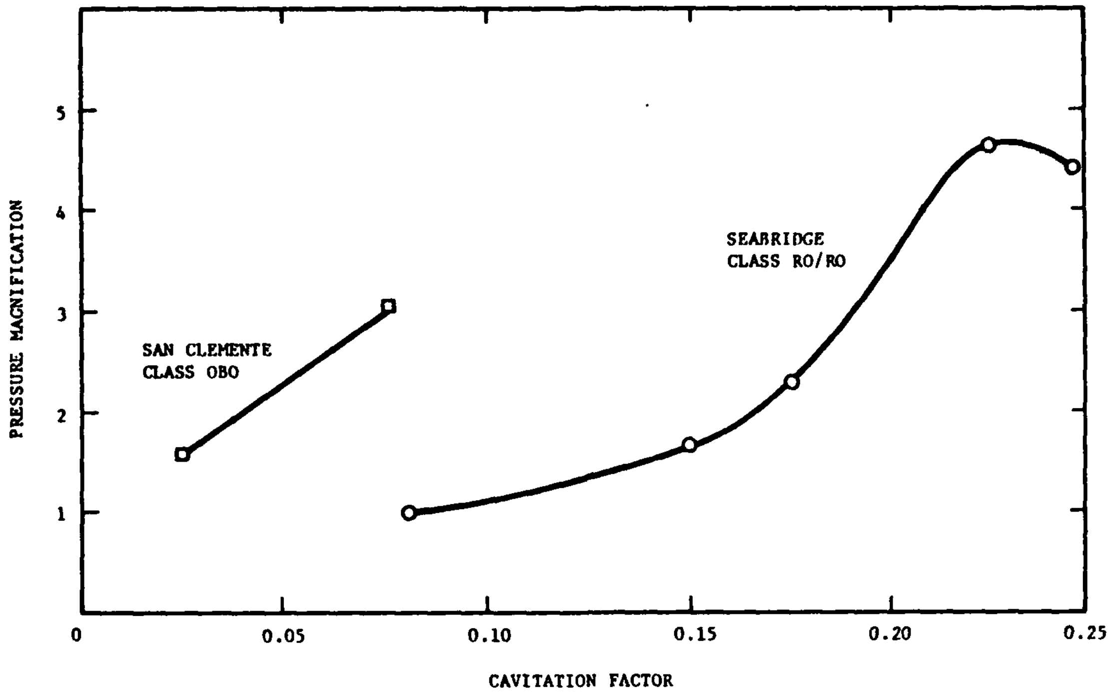


FIGURE 36. RATIO OF MEASURED HULL PRESSURES WHERE CAVITATION EXISTS TO CALCULATED, NONCAVITATING PRESSURES

TABLE 21. HULL FORCES AND MOMENTS DUE TO CAVITATION EFFECT

RPM	F_x (lb) (Calculated)	M_y (ft-lb) (Calculated)	Cavitation Factor	F_x (lb)	M_y (ft-lb)
106.7	5006	33,071	3.00	15,000	100,000
100.0	4398	29,048	4.20	18,470	122,000
95.0	3969	26,216	4.25	16,870	111,420
90.0	3562	23,530	2.37	8,442	55,760
85.0	3177	20,987	1.93	6,132	40,505
80.0	2814	18,591	1.54	4,334	28,630

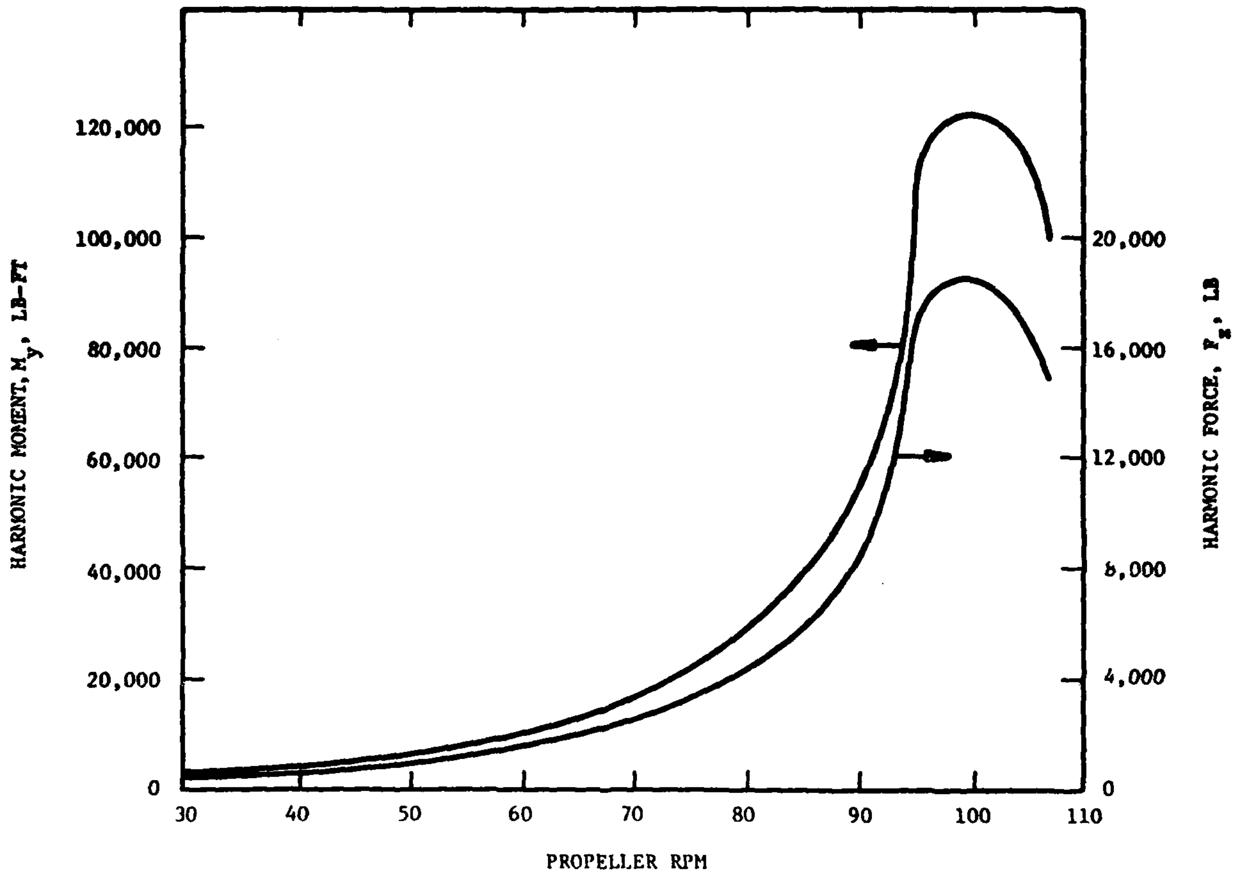


FIGURE 37. VERTICAL HARMONIC FORCE AND TRANSVERSE BENDING MOMENT GENERATED ON HULL BY THE PROPELLER

2.3.16. Determine Forced Lateral Response of Shafting (Rigid Hull)

and

2.3.17. Determine Forced Lateral Response of Shafting (Flexible Hull). Although these estimates should be made in the course of the design development, in this demonstration example the step is bypassed (as it was in the original work), but is included in Block 21.

2.3.15. Determine Forced Response of Machinery Space

and

2.3.18. Conduct Superstructure Modal Analysis. These two steps in the development program are combined in a modal analysis and for d response calculation, which includes the machinery space and superst' :ture as one common substructure. In the course of a design development, it would be desirable to analyze these two sections separately and to combine them as substructures in the final finite-element model. However, in this example, it was considered advisable in the interests of obtaining good correlation with experimental results to combine both the machinery space and the superstructure into one large substructure since the two would be used in a beam model rather than a large finite-element model.

In the original analysis of the ship in 1969 and 1970, the machinery space was analyzed with a coarse STARDYNE model to give inputs as sprung masses to the general beam model of the ship. This model extended to the main deck, but did not include the propulsion system. The model gave indications of the type of difficulty that was experienced, but these were not recognized at the time. In more recent finite-element analyses, the modeling has been much more realistic, but there has never been a good opportunity to compare predictions with measurements on the ship. To support the belief that good finite-element analyses will predict potential vibration problems, it was decided to make the analysis on this ship to the standard that has been applied on other ships that have been analyzed in recent years. To further assure this, it was decided to include the superstructure with the machinery space model since they are closely coupled. Of course, the propulsion system should be included. Since the water inertia values that have been used in past analyses have always been questionable and are a significant part of the total mass, it was decided to include a new representation of water inertia recently conceived and not developed. Because, among the possible computer programs for analyzing the structure, the ANSYS computer program has an element that appeared to be directly applicable to the introduction of water inertia under the new concept, and because of the good experience of SwRI with ANSYS, it was decided to use this program for the analysis. This meant developing a familiarity with a new program instead of NASTRAN, which LR&EC has customarily used for large structural vibration problems. It has subsequently been determined that the water-inertia matrix could be introduced into the NASTRAN program. After a long period of frustration, it was discovered that the ANSYS initial introduction of water inertia was in error so that the choice of computer program system for this example was unfortunate. The error in the water-inertia matrix was subsequently corrected, however.

In subsequent sections, the following topics will be discussed:

a. The Structural Model
b. The Treatment of Water Inertia
c. The Natural Frequencies and Mode Shapes
d. The Response to Axial Excitation at the Propeller
e. General Discussion of the Substructure Model

(a) The Structural Model

The section of the ship modeled in this analysis is the machinery space bounded by its forward and aft bulkheads, which are assumed to be fixed. Included in the model are the complete superstructure, which generally falls within the length bounds of the bulkheads, and the propeller and shafting, which is free to move axially in its bearings and is coupled to the machinery space at the thrust bearing. An inboard profile of the region is presented in Figure 38. A typical section is shown in Figure 39, and Figure 40 shows the after engine room bulkhead at Frame 181. These sections contain the grid points used to define its structure and mass loading. In the machinery space length extending from Frame 157 to Frame 181, there are five (5) of these web frames, which with the bounding bulkheads give seven (7) longitudinal subdivisions. There are nine (9) deck levels in the machinery space and seven (7) in the superstructure. A total of 577 grid points are used to define the structure. These carry masses and are connected by membranes (plates whose bending is neglected), spars (columns whose bending is neglected), and beams. A total of 2595 elements, including masses, were used. The computer program determines the mass of the structure, but additions must be made for equipment, furnishings, electrical and piping fittings, machinery, water inertia, etc.

The computer program assembles all of these structural elements and computes their assembled stiffness and mass matrices, which have generally at least three times as many elements as the number of grid points. These matrices are difficult to analyze, so the stiffness-mass matrix is customarily reduced to an equivalent dynamic matrix referred to grid points assigned by the analyst as capable of representing the dynamic behavior of the structure. For this model, there are 72 dynamic-degrees-of-freedom. The choices of degrees of freedom represent what the analyst expects to be good locations to represent the structure and define the vibration patterns. To reduce cost and complexity, he likes to keep the number small, but by so doing he may fail to reflect some particular vibration pattern that he did not anticipate. Because all the vibration that was strongly observable in the machinery space was symmetrical, i.e., the same on both the port and starboard sides, only the symmetrical vibrations have been considered in this study. Good practice in studying ship hull vibration would dictate that both the symmetric and anti-symmetric vibrations be analyzed. Once a model for symmetric vibrations has been set up, it is only necessary to change boundary conditions, repeat the last calculation, and interpret the results to cover the anti-symmetric vibrations. Anti-symmetric vibrations are not as common as symmetric, but they are far from unusual.

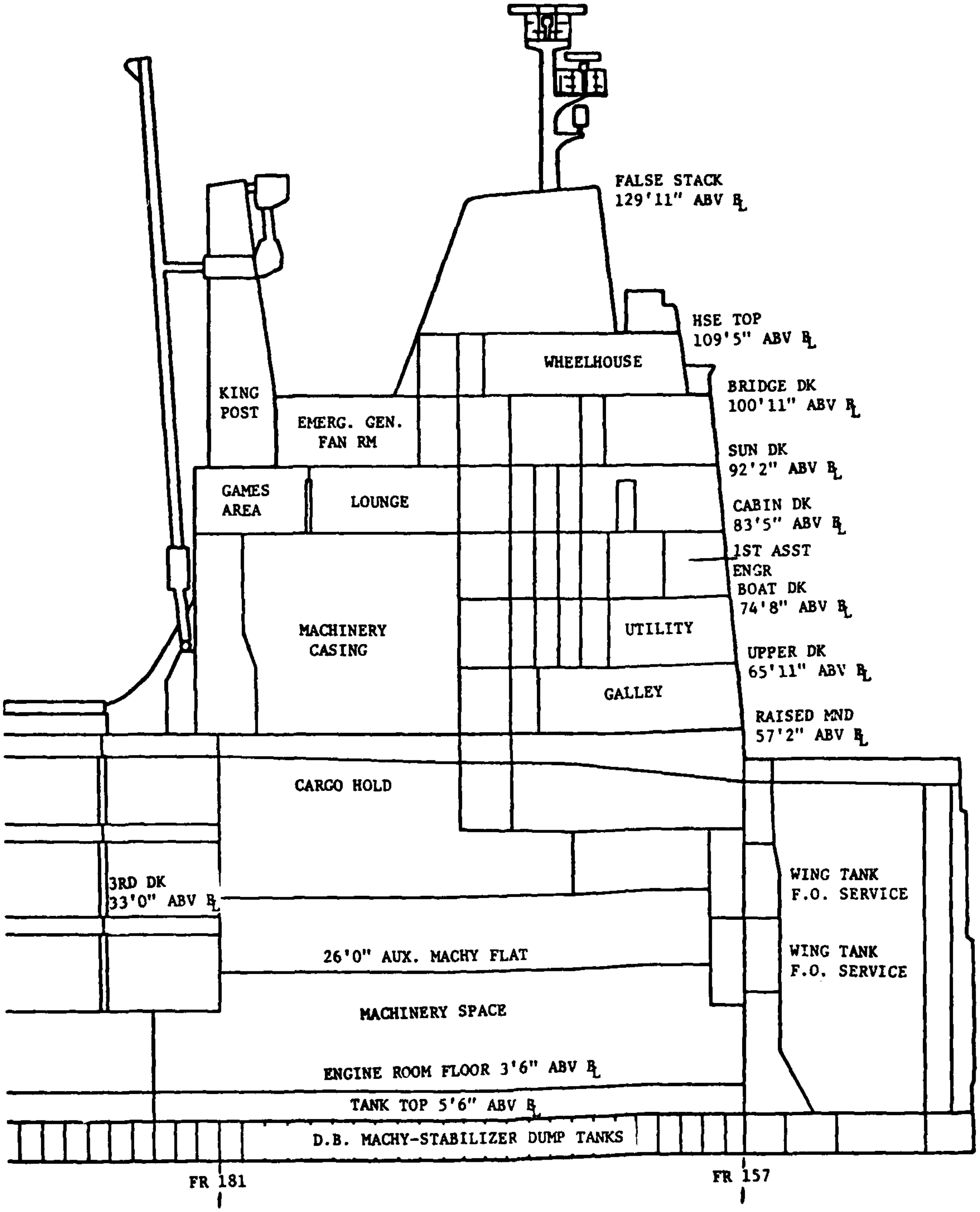


FIGURE 38. INBOARD PROFILE

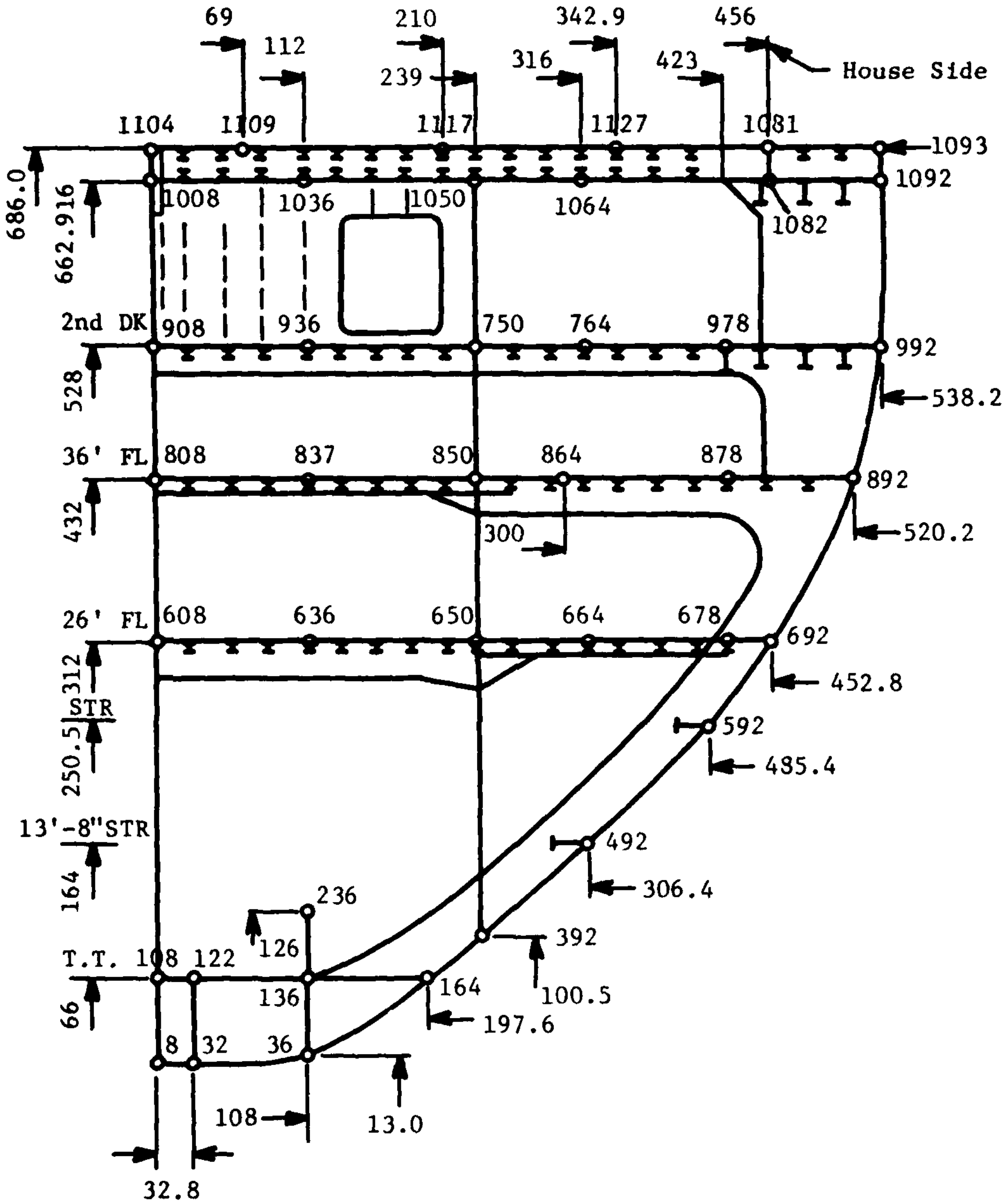


FIGURE 39. GRID POINTS ON FRAME 170,
X = 396 IN.

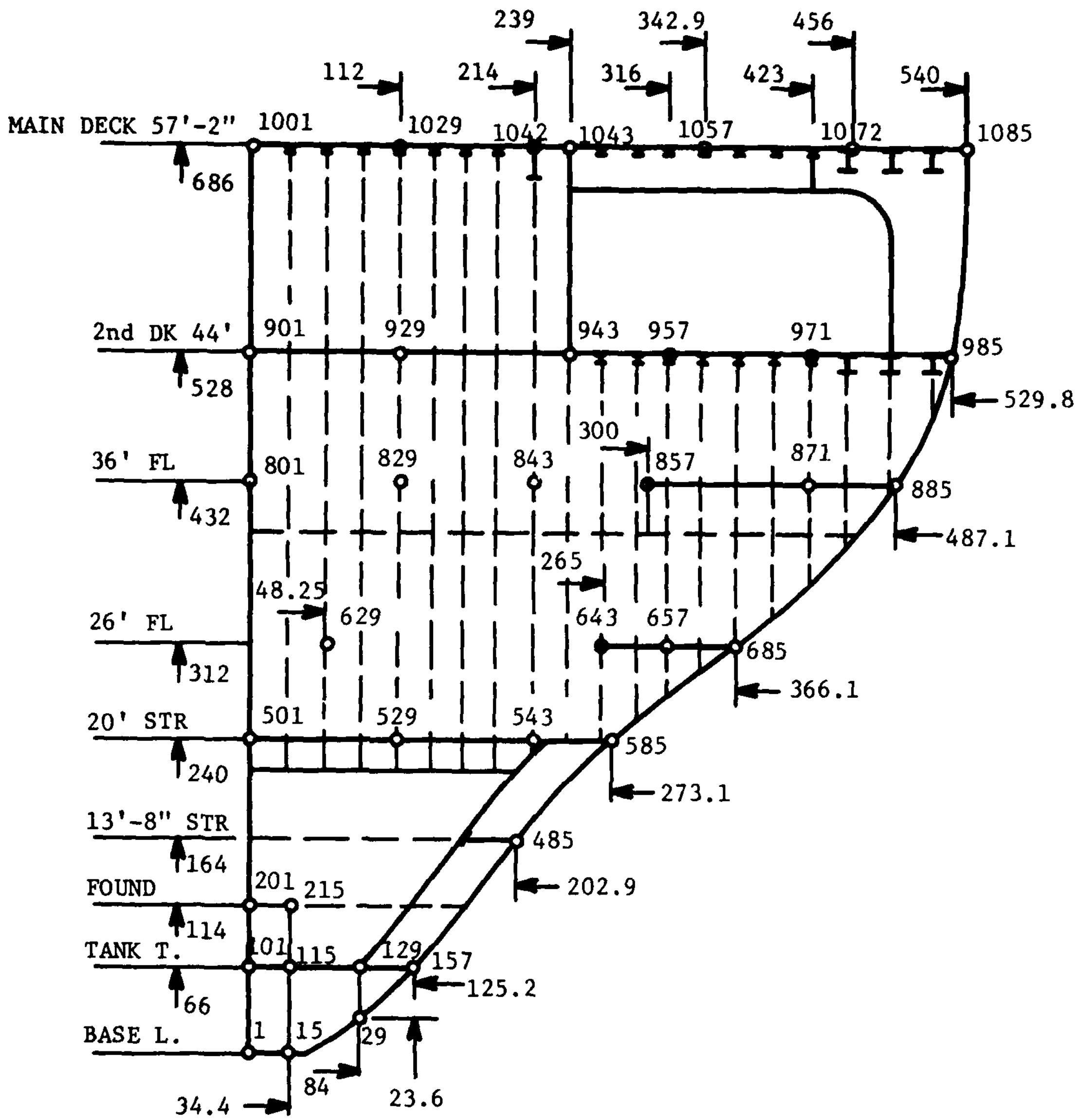


FIGURE 40. GRID POINTS ON FRAME 181,
X = 0 IN.

The model was loaded at the propeller with longitudinal excitations. Because the hull substructure did not extend aft to the propeller, no horizontal or vertical excitations on the propeller or hull are included in this analysis.

(b) Water Inertia

One of the present serious deficiencies of the analysis of structures in water by use of finite elements is the treatment of the effective mass of the water moving with the structure. This same difficulty plagued early efforts to predict the natural frequency of ships as beams until F. M. Lewis and J. Lockwood Taylor developed procedures for including the water mass.

In a structure, it is desirable to assign one mass to each grid point in contact with the water. However, this is not enough because motion at one location will generate pressures at adjacent locations. One way of representing the water accurately is to establish a volume of grid points in the water and use fluid elements to develop the pressures in the water and on the ship from the vibratory motion of the ship. There are a limited number of cases where this has been done, and the results are good. However, the procedure adds complexity to a computer analysis that is already large.

For this analysis a procedure for computing the mass matrix of the water inertia over a number of grid points on the bottom was applied. It is based upon the pressure generated on a flat surface (an extension of the pressure on the surface of a sphere) by a small vibrating piston located in the surface. The pressure is proportional to the area and the vibratory acceleration of the exciting piston and inversely proportional to the separation between the exciting piston and the location of pressure measurement. Thus, the mutual force between small areas A_m and A_n is

$$F_{mn} = \frac{C A_n A_m a_m}{r_{mn}}$$

where

C = a factor reflecting the density of the fluid and the proportions of the pistons

A_m, A_n = the areas associated with grid points m and n

r_{mn} = the minimum surface distance between grid point m and n on the ship's hull

a_m = the acceleration of the piston m

Clearly the quantity $\frac{C A_m A_n}{r_{mn}}$ has the units of mass, and so the force/acceleration interaction between grid points on a surface bounding a fluid can be represented by a matrix of mass terms.

There are corrections for the physical dimensions of the sending and receiving pistons. Where only half a surface is modeled, the interaction with the other half can be represented for symmetrical and antisymmetrical motions by dummy grid points. The influence of free water surface can also be represented by using a mirror of grid points above the water surface.

If the dimensions of the pistons are small compared to the wave length of the transmitted vibration, the pressure wave is transmitted as a spherical wave and there is no significant correction required for the location of the receiving piston relative to the sending piston.

It was found that the mutual interaction between pistons is significant for considerable distance such as for the length of the machinery space. Where the distance is long, there can be a phase shift between the force at the remote piston relative to the acceleration at the transmitting piston. The finite-element model of the fluid normally would not include this.

A paper is being prepared for submission to the Journal of Ship Research covering the background and details of this water-inertia treatment. In the machinery space model, it was used in its simplest form with minimal corrections for piston dimensions relative to the separation of the sending and receiving pistons.

(c) Natural Frequencies and Mode Shapes

The modal analysis of the stiffness and mass matrices representing the structure, when reduced to equivalent matrices having 72 degrees of dynamic freedom, gave the following frequencies and general mode shapes.

Mode 1: Frequency = 1.9 Hz

The maximum displacement is a vertical motion on the 26-ft flat on the centerline at Frame 165. The motion drops rapidly in adjacent points. This grid point is on top of a portable plate and should not have been selected as a degree-of-freedom point. The selection was made because it was known that on trials the 26-ft flat vibrated heavily.

Although the 26-ft flat is a light structure and vibrates noticeably, this particular vibration mode is probably spurious and arises because the weights on the 26-ft flat were distributed evenly among the grid points rather than concentrated at the particular items of equipment. It will be shown later that this particular mode is not excited by the longitudinal force at the propeller.

Mode 2: Frequency = 6.2 Hz

The maximum displacement is still on the 26-ft flat and still at Frame 165, but in this case at a grid point 214 in. (17 ft 10 in.) off the center. The structure at this location is permanent, but light, and would not be loaded as heavily as is done by making it a dynamic degree-of-freedom. The frequency is therefore probably spurious.

Mode 3: Frequency = 6.5 Hz

The maximum displacement for this mode is still on the 26-ft flat at Frame 165, 214 in. off the center. However, the vibration is much more widespread with amplitudes in the superstructure, along the lower engine room stringer, and axial motions in the shaft one-eighth as large as the maximum amplitude. This is probably the lowest real natural frequency.

Mode 4: Frequency = 7.1 Hz

The maximum displacement for this mode occurs in the lower engine room stringer at Frames 174 and 177. Reference to the forced vibration response to the axial force from the propeller indicates that this mode is not strongly excited by the longitudinal propeller force. However, it might be excited by vertical and longitudinal forces and moments at the propeller. The kingpost on the after side of the superstructure appears to show fore and aft resonances at this frequency.

Mode 5: Frequency = 8.7 Hz

The maximum amplitude of vibration is at the main control console on the 26-ft flat. There also appears to be some motion on the 36-ft flat in the same general region.

Mode 6: Frequency = 9.2 Hz

The maximum motion at this frequency is a lateral motion of the kingposts This does not appear to be strongly excited by the axial propeller force, but might be excited by lateral and vertical forces.

Mode 7: Frequency = 10.2 Hz

The maximum motion associated with this mode occurs on the 36-ft flat, Frame 170, centerline at the main control console. There is also significant motion in the framing in the lower engine room at Frame 170. This mode is quite strongly excited by the axial propeller force.

Mode 8: Frequency = 10.8 Hz

This is the primary mode of the whole structure vibrating vertically against its restraints at the forward and aft bounding bulkheads. The motion is accentuated at the 36-ft flat and in the kingposts. It includes bending in the side frames of the machinery space. As will be shown later, this mode is excited by the axial propeller force. It might be more strongly excited by vertical forces and moments about a transverse axis at the propeller and on the hull.

Mode 9: Frequency = 12.3 Hz

The location of the point of maximum motion is again on the 36-ft flat, Frame 170, centerline. The motion is vertical and although

the relative motion of the 36-ft flat is larger than for the previous natural frequency (Mode 8), the remaining motions in the ship are smaller.

Mode 10: Frequency = 12.4 Hz

This is the primary mode of interest in which the propeller and shafting vibrate against the rest of the structure. This mode is strongly excited by the propeller.

Mode 11: Frequency = 13.6 Hz

The mode shows the superstructure vibrating against the lower ship side with small shaft motion.

Higher frequency modes are above 15 Hz and outside the range of interest. These modes will be discussed further in section (e) after consideration of the response of the structure to forced vibration from the propeller.

(d) Response of Substructure to Axial Harmonic Forces at Propeller

Using the substructure as modeled by 72 dynamic degrees-of-freedom, the response to an axial force at the propeller was determined. To avoid confusion from spurious resonances, a moderate amount of hysteretic damping was introduced (3 percent). Some of the results are plotted in Figures 41 through 47.

(e) General Discussion of Substructure Model

In these calculations, the substructure is taken to be supported rigidly at the forward and aft engine room bulkheads, which are Frames 157 and 181. Note from Figure 38 that the bulkhead at Frame 181 does not extend to the tank top. Because the substructure in reality is supported by the elastic ship rather than the assumed rigid bulkheads, the natural frequencies in which the whole structure participates will probably be lower than predicted by these calculations. Also because of the flow of vibratory energy from the machinery space into adjacent structure (and also incomplete damping), the predicted amplitude of motions arising from the longitudinal excitation from the propeller will be high. Therefore, Figures 41-47 indicate relative vibration amplitudes throughout the structure and are not the amplitudes predicted for the complete ship. The excitation from the propeller and on the hull in the vertical and horizontal planes is not considered since it acts outside the substructure. Many modes which are weakly coupled to the propulsion shaft system will be more important than the vibration excited by the axial forces at the propeller. All of these effects are included and combined with the complete ship in Blocks 21 and 22.

The vibration analyst would probably interpret the information that has been generated as follows:

(1) There is a strong longitudinal vibration in the shafting system. Its frequency at 12.4 Hz is about 17 percent above the maximum propeller excitation at 10.6 Hz. However, it is a strong vibration with

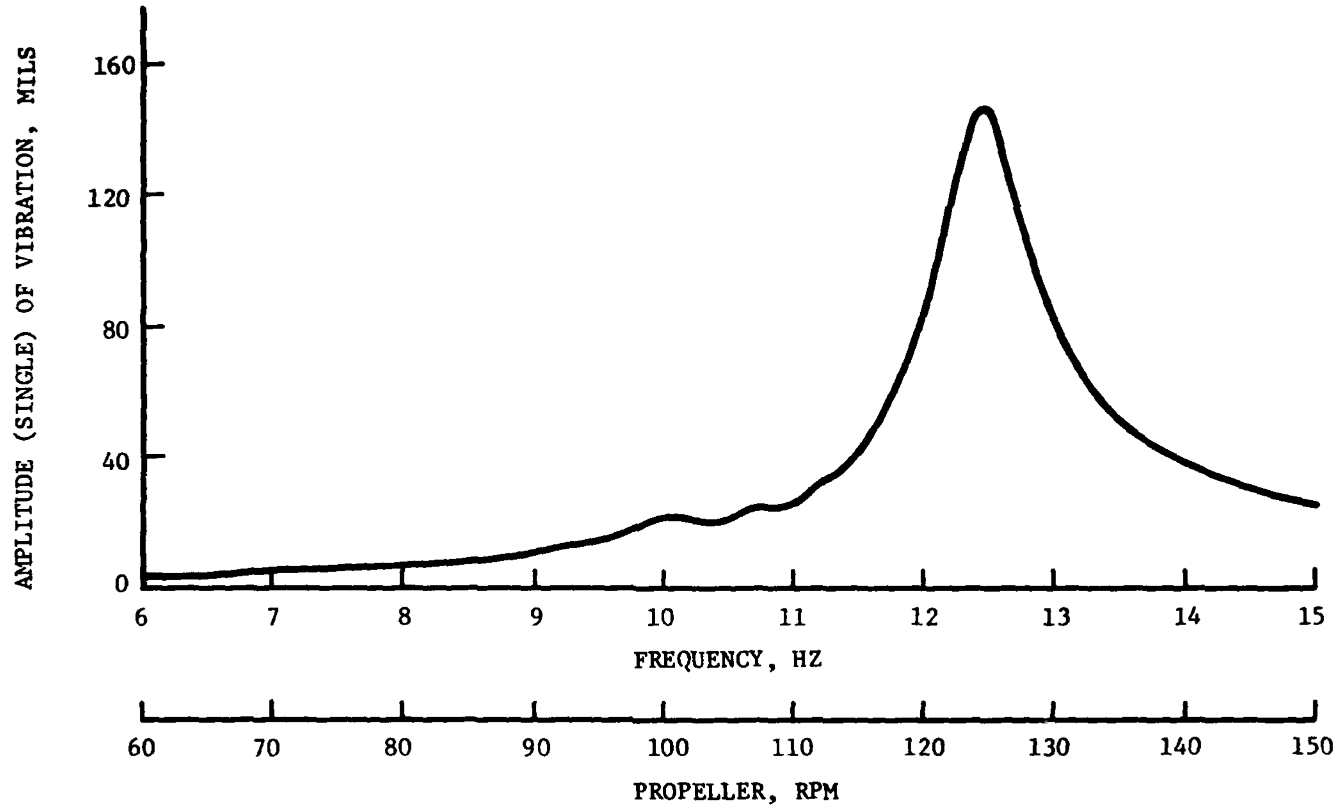


FIGURE 41. AMPLITUDE OF AXIAL MOTION AT THE PROPELLER

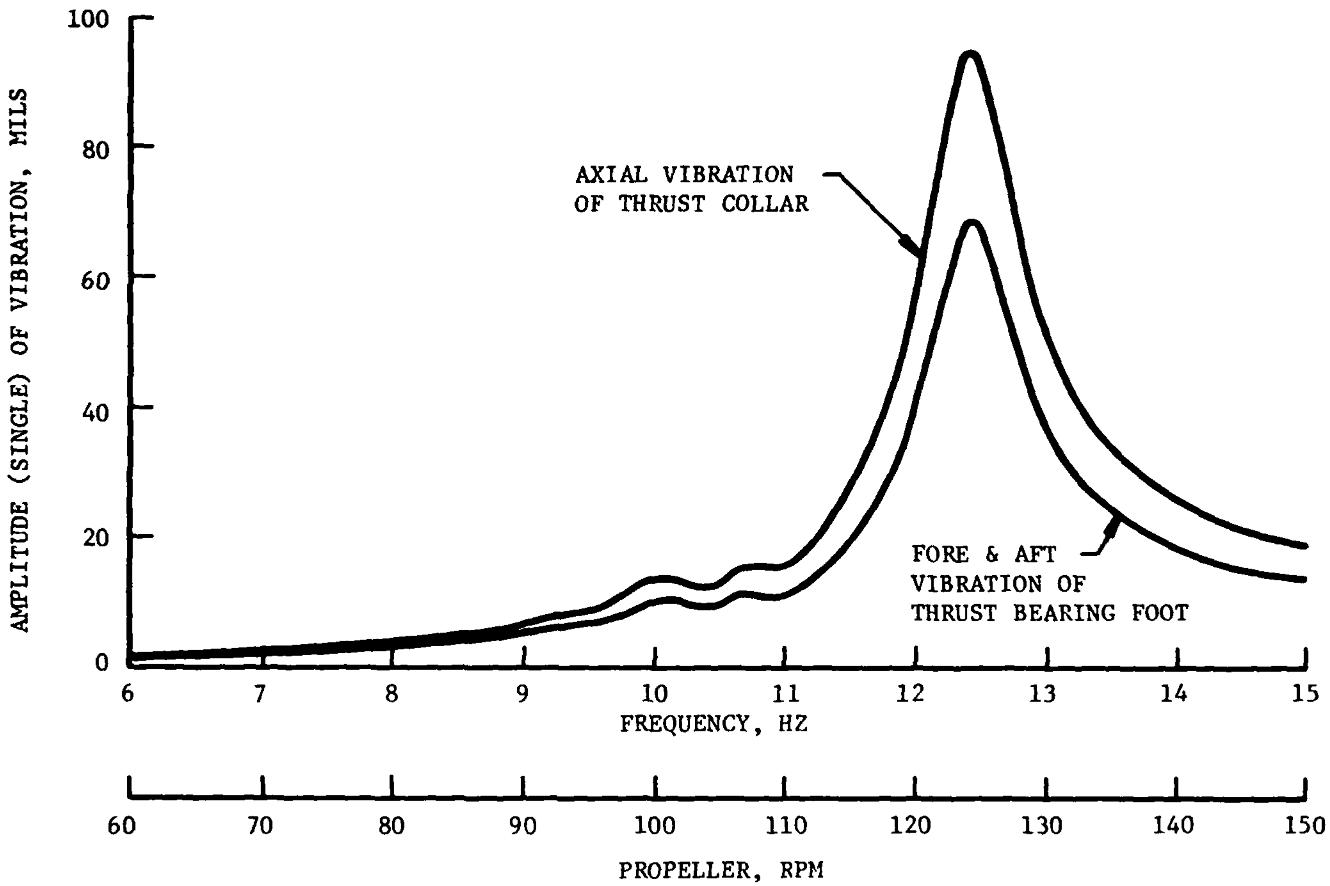


FIGURE 42. AMPLITUDE OF FORE AND AFT VIBRATION AT PROPULSION SHAFT THRUST COLLAR AND THRUST BEARING FOOT

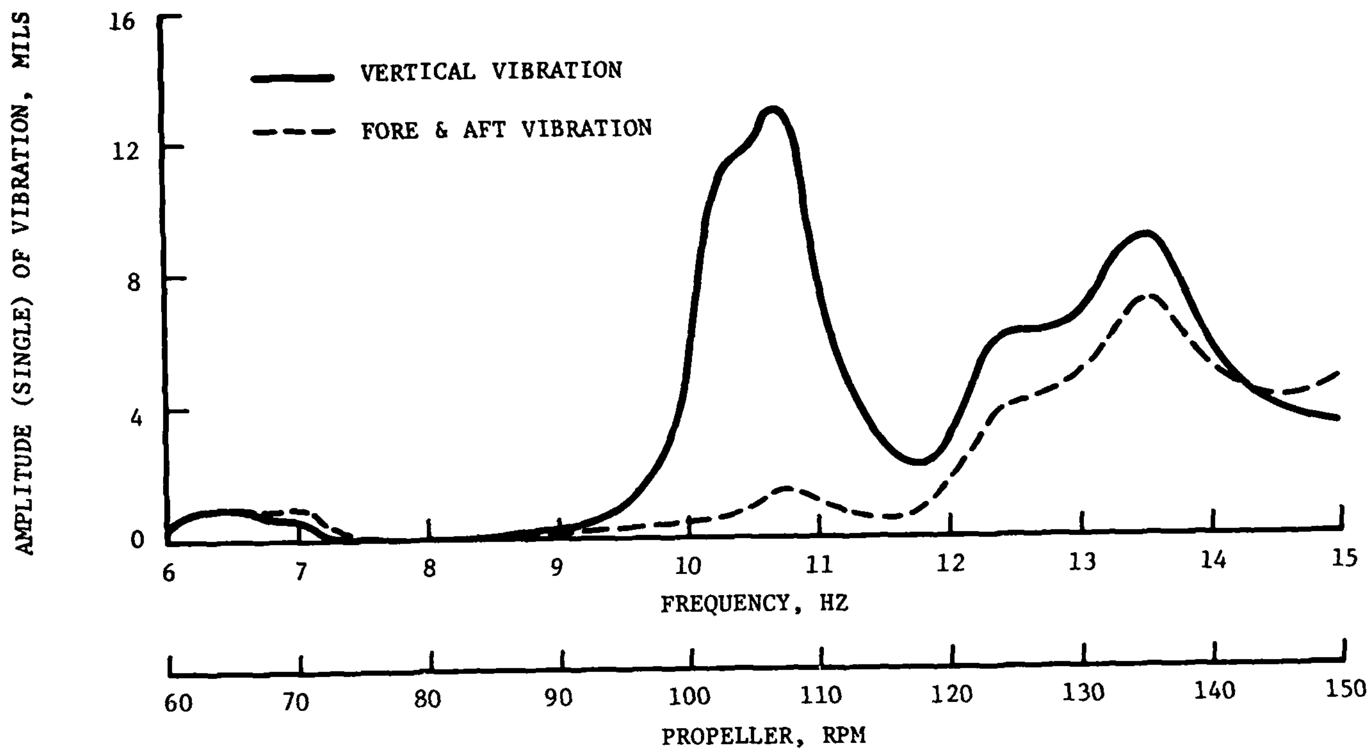


FIGURE 43. VIBRATORY MOTION ON BRIDGE, FRAME 164 AT CENTERLINE

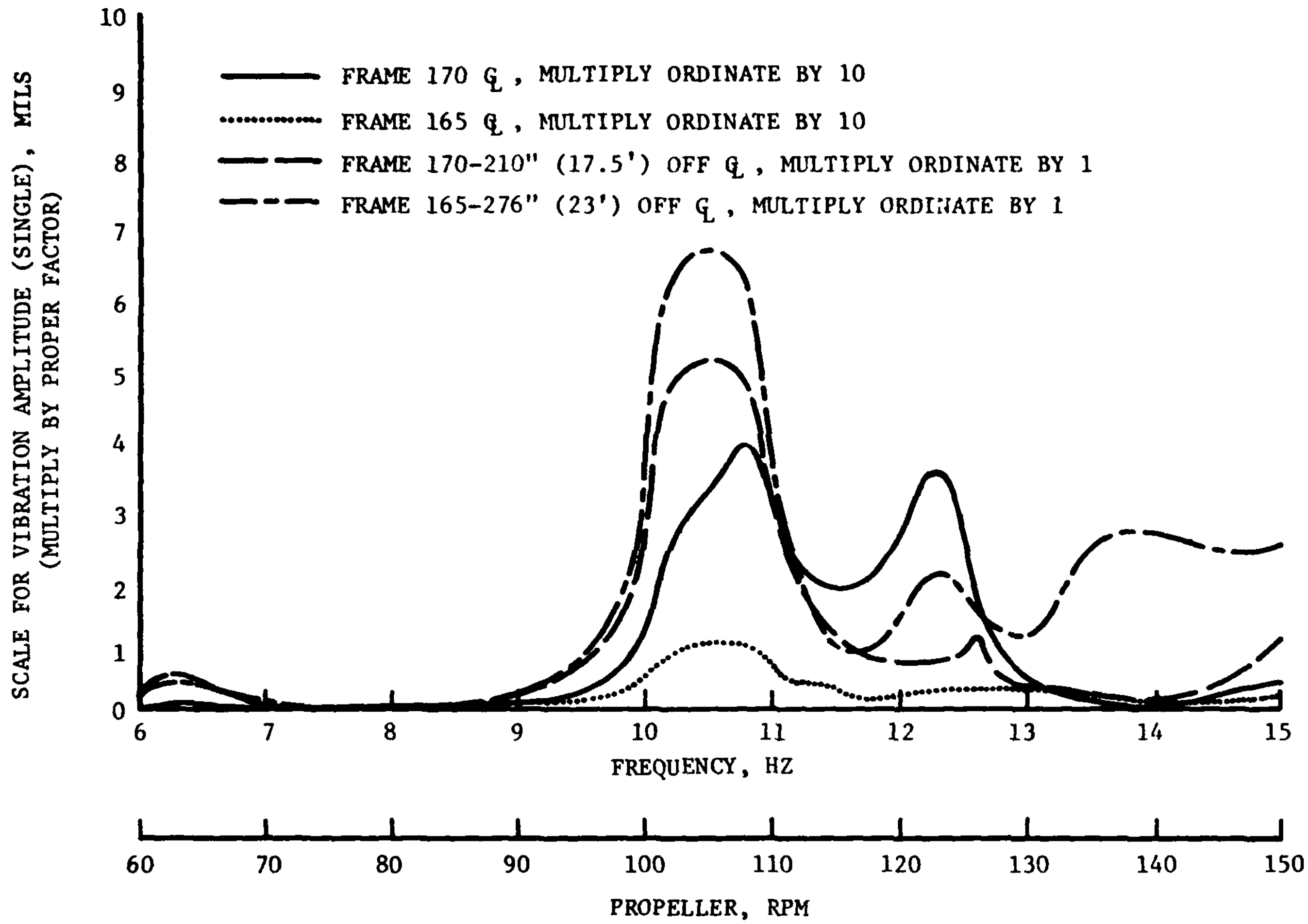


FIGURE 44. VERTICAL VIBRATION ON 36-FT FLAT GENERATED BY AXIAL PROPELLER FORCE

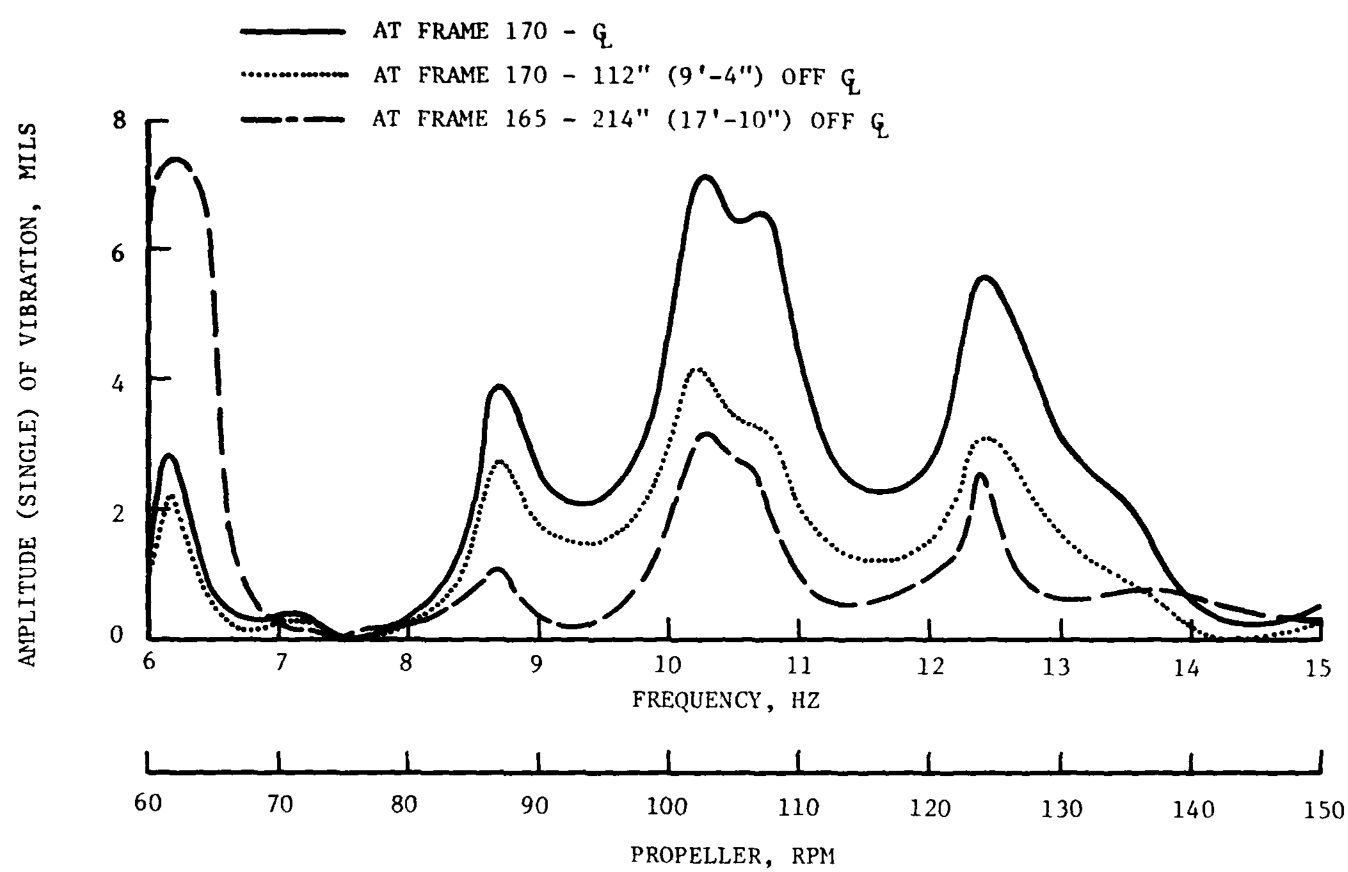


FIGURE 45. VERTICAL VIBRATION ON 26-FT FLAT EXCITED BY AXIAL HARMONIC FORCE AT THE PROPELLER

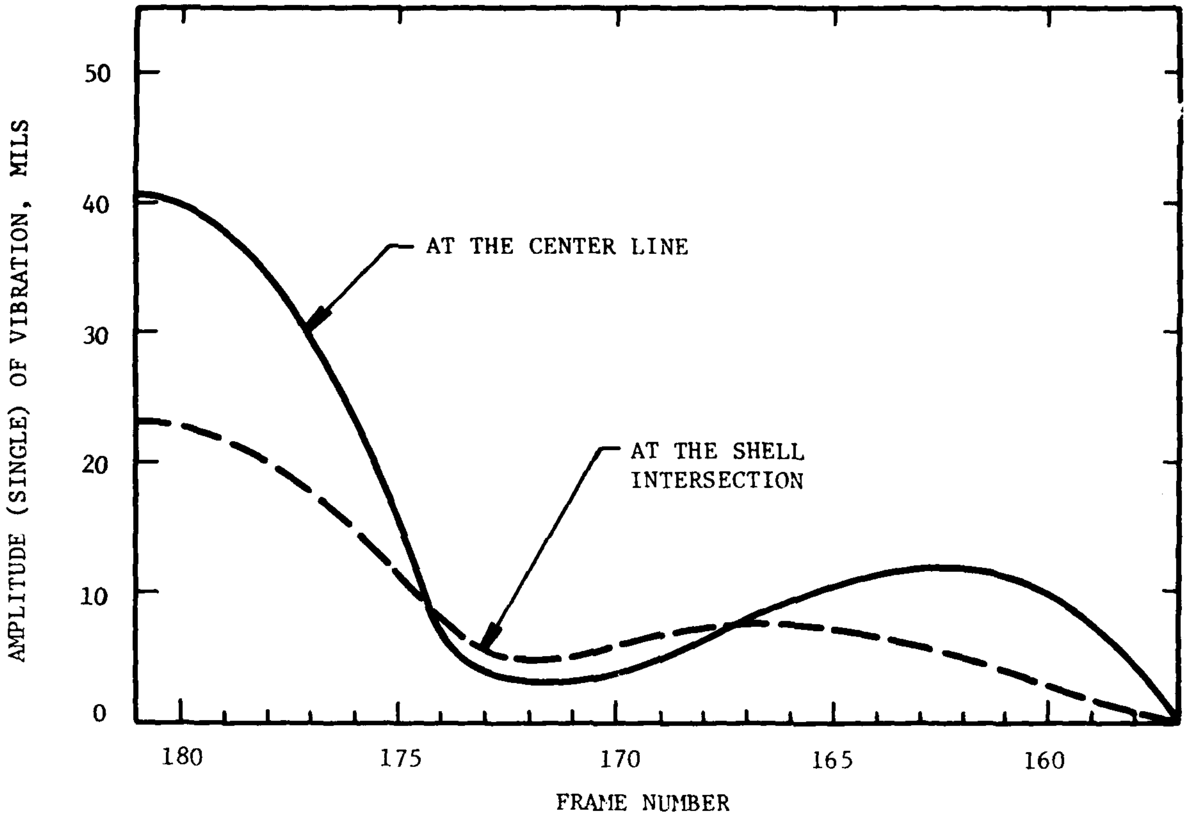


FIGURE 46. VERTICAL VIBRATION IN TANK TOP DUE TO LONGITUDINAL EXCITATION AT THE PROPELLER AT 12.43 HZ (Frequency of Longitudinal Shaft Resonance)

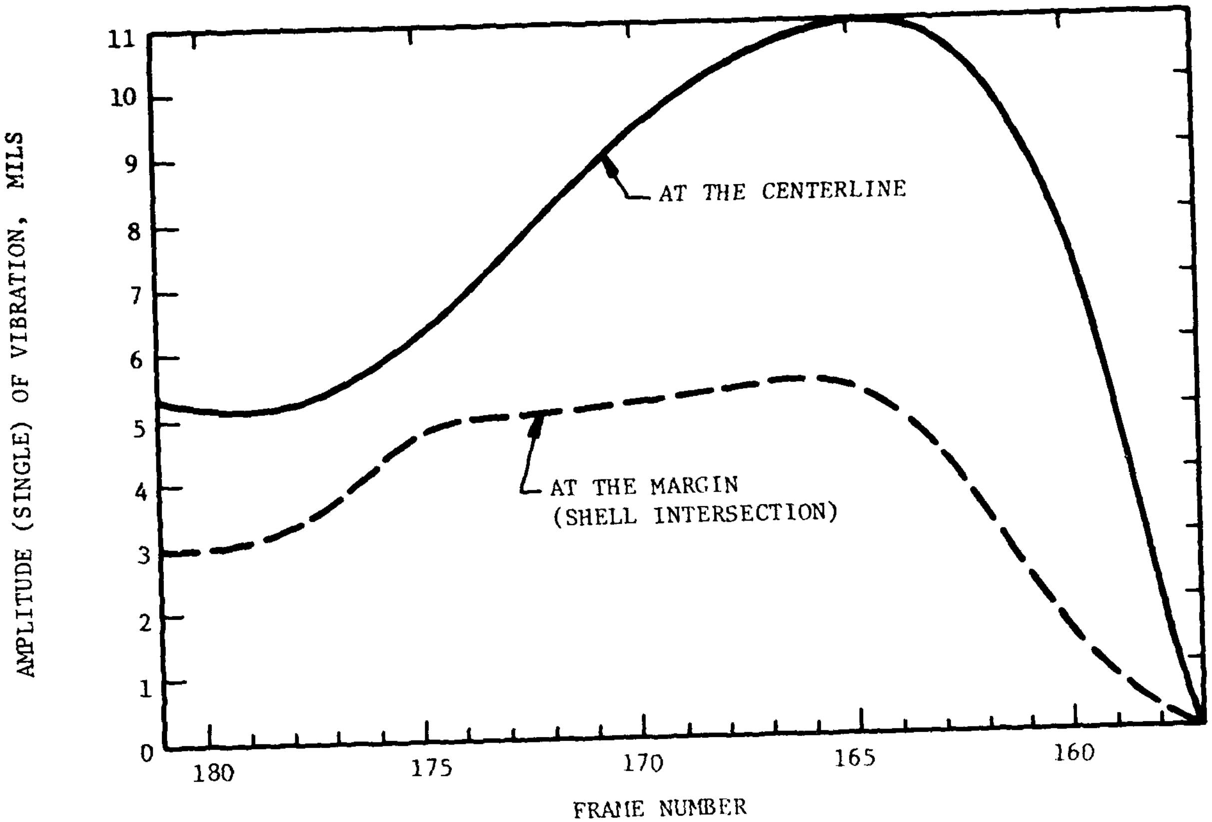


FIGURE 47. VERTICAL VIBRATION IN TANK TOP DUE TO LONGITUDINAL EXCITATION AT THE PROPELLER AT 10.8 HZ (Frequency of Superstructure Resonance)

broad flanks, and its frequency will drop when introduced into the elastic ship. It is lower than would have been expected from the preliminary calculation of Block 4, which would indicate that the foundation structure has too low a stiffness. This is borne out by Figure 46, which indicates a large slope in the deflection curve of the bottom. It would appear that an effective way of improving the stiffness would be to make the transverse structure between Frames 177 and 181 more rigid.

(2) The natural frequency of the whole engine room and superstructure, Figure 43, falls at just about maximum speed. This frequency will drop somewhat when incorporated in the elastic ship, but even so can be expected to give some vibration difficulties. The amplitudes generated by the longitudinal propeller excitation will probably be acceptable, but this mode would be more strongly excited by propeller and hull force and moments acting in the vertical plane. Raising the frequency above the operating speed may be difficult when one recognizes that the predicted value at this stage will drop when the substructure is incorporated in the elastic hull. If raising the frequency appears desirable, it should be accomplished by increased transverse stiffness at about Frame 165. See Figure 47. The possibility of deepening the transverse bulkhead between the 36-ft flat and the second deck to between the second and main decks might be explored. It might also be well to make the web frame at Frame 165 heavier.

(3) The natural frequency and mode-shape determinations show large motions on the 26-ft and 36-ft flats. In particular, the 26-ft flat that is the location of the Engine Control Console is a region where freedom from vibration is critical. It would be advisable to improve the finite-element model in these regions and to give some consideration to structural modifications to lessen the vibrations.

2.3.19. Determine Natural Frequencies and Forced Response of Rudder. This study was not made in the original work and is not done in this program. Generally rudders, even when in the air, have a low transverse natural frequency and thus generally have little influence on the hull vibration.

2.3.20. Evaluate Local Plating Design. This important aspect of good vibration analysis has not been treated in this program. This aspect of the vibration analysis can be handled most realistically if performed after the general vibration level in different regions of the ship is determined in Block 23.

2.3.21. Assemble Model of Entire Ship. It was the intention of this program to carry the analysis through to a prediction of vibration in the ship and to compare the predictions with measurements. The cost and time of running the finite-element analyses of the Machinery Space and Superstructure have precluded this.

Under this item, the finite-element analysis of the machinery space and superstructure would be represented as a subsystem, System 4 in the original ship model as shown in Figure 13. This subsystem would be a beam with several sprung masses and would include the shafting system and longitudinal excitation at the propeller. With this System 4

included, the whole ship would be properly represented as a damped, mass-elastic system excited by five components of propeller force and moment (torsional moments about the rotational axis are omitted) and six components of hull force and moment.

Although this ship is too complex and the excitation frequency too high to give good results from a beam model, this model was an improvement on the procedure in general use when it was developed in 1970. It is expected that the results would be reasonably confirmed by the trial measurements since the critical regions of the machinery space and superstructure are accurately modeled. The connecting ship structure transmits forces and motions to this region and dynamically interacts at the boundaries. The assumption is that the accuracy of these boundary reactions will not too strongly influence the response in the critical regions.

2.3.22. Determine Vibration Amplitudes and Stress Levels of Complete Ship. As discussed under Section 2.3.21, it has not been possible to accomplish this.

2.3.23. Conduct Shaker Tests. A shaker test was made to determine the response at the thrust bearing without the propeller and shafting connected. Although this test could be used to verify the analysis procedures, the data obtained were limited, and a modified substructural model would be necessary to make the analytical study for comparison with the test data. These tests will be considered under Block 27.

2.3.24. Assess Local Vibrations, Structural Damping, and Modeling Techniques. No work has been done in this design block.

2.3.25. Measure Vibrations During Sea Trials

and

2.3.27. Compare Measured Vibration with Calculations. During sea trials, the ship was observed to have heavy vibrations of the following types and locations:

(a) Longitudinal vibration of shafting and thrust bearing

(b) Vertical vibration on the 26-ft and 36-ft flats

(c) Vertical vibrations in the cabin deck lounge over the machinery casing

(d) Vertical vibration on the bridge.

Graphs of the data obtained in exploring these vibrations are presented in Figures 48 through 53. Figure 48 shows that the peak amplitude of fore and aft vibration had probably not been reached at 102 RPM. In Figure 49, the twice-blade frequency fore and aft vibration is plotted in an effort to determine the natural frequency of longitudinal vibration. The results are not very conclusive since the excitation at the low speed required to define the resonance is low. However, the combination of amplitude and phase shift (the phase is with respect to an arbitrary angle)

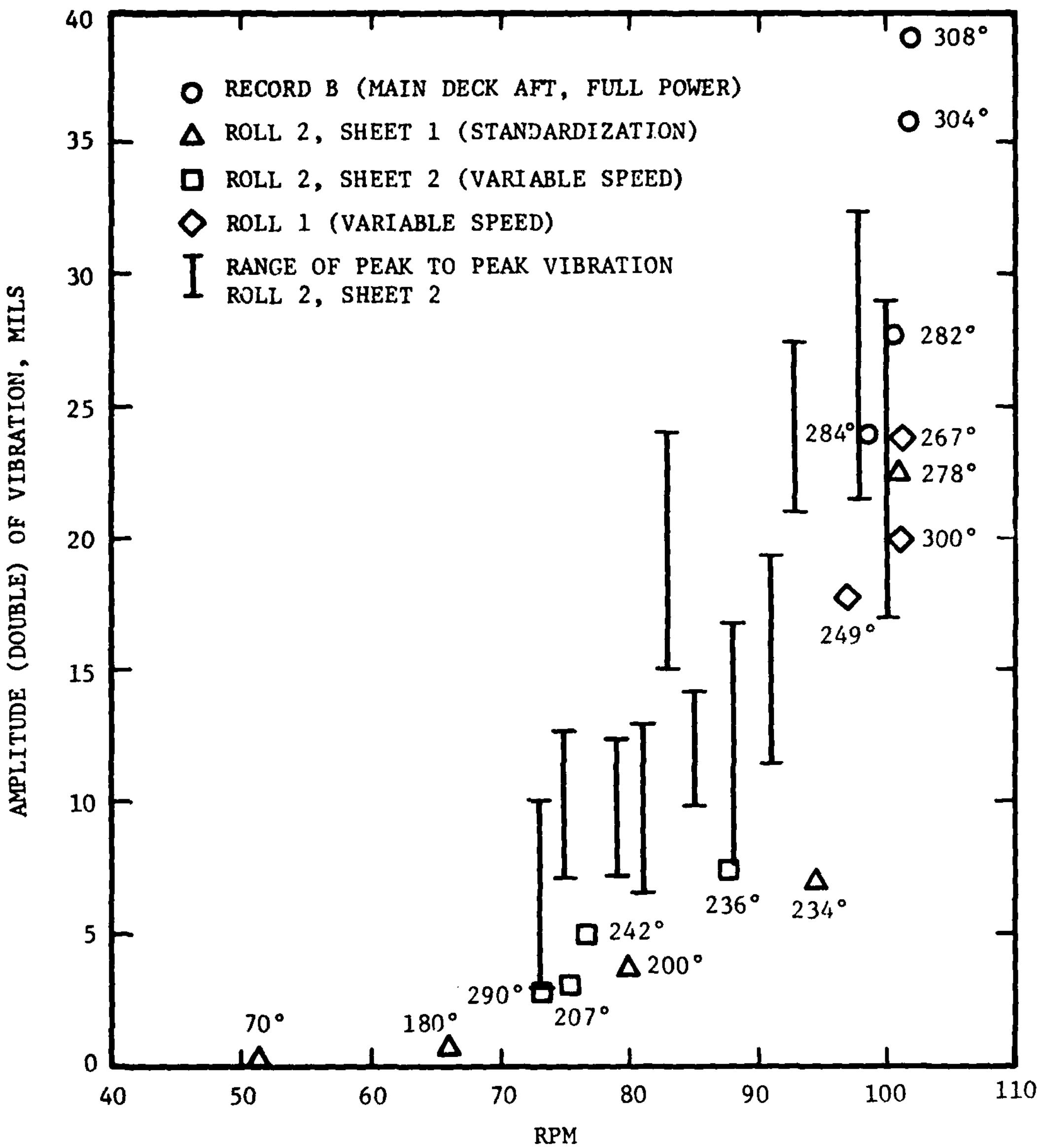


FIGURE 48. DOUBLE AMPLITUDE OF SIXTH ORDER, FORE AND AFT MOTION OF THRUST BEARING FOUNDATION

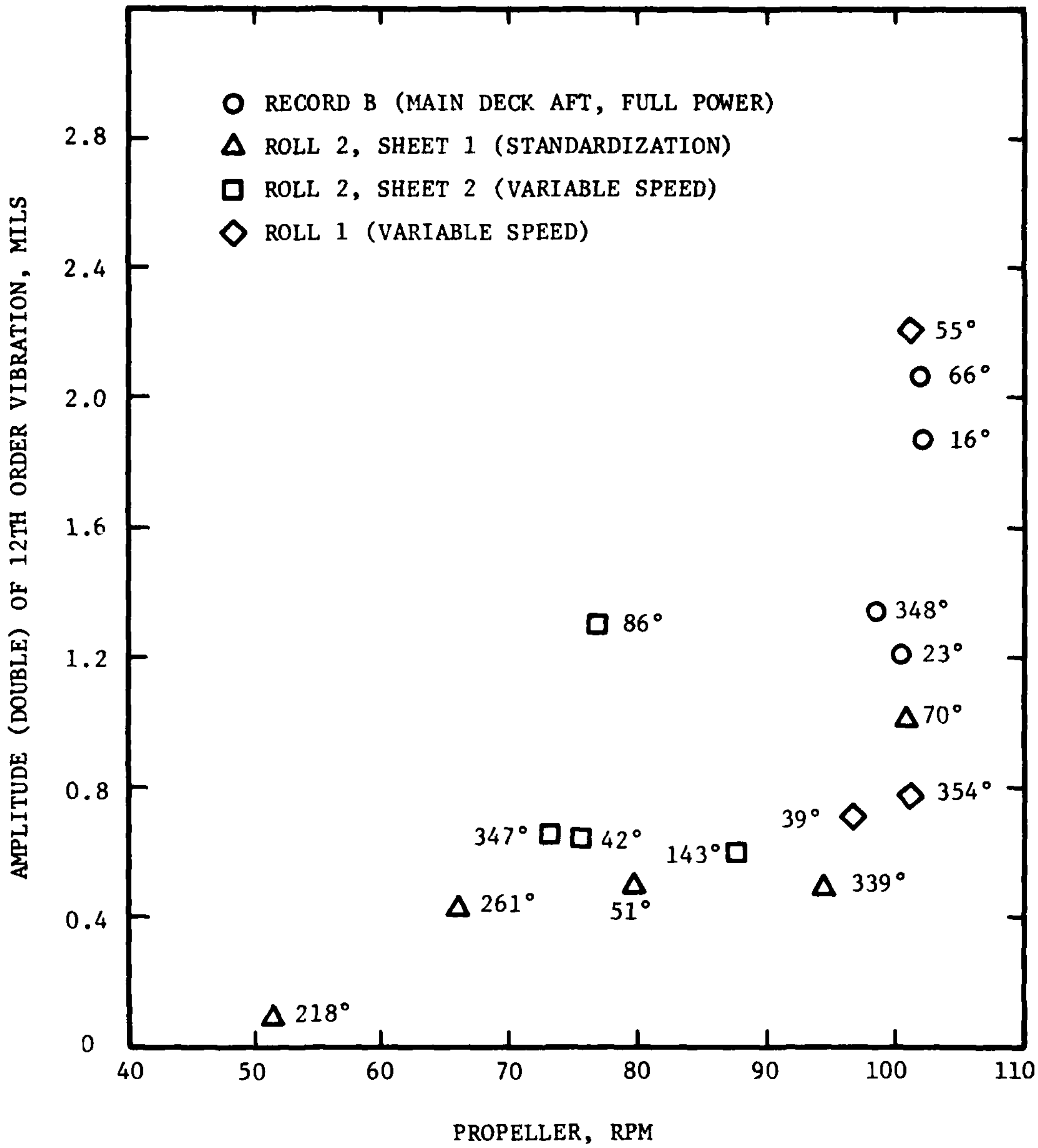


FIGURE 49. DOUBLE AMPLITUDE OF TWELFTH ORDER, FORE AND AFT VIBRATION ON THRUST BEARING FOOT

might be construed to indicate a resonance at about 75 RPM or 15 Hz, or possibly at 14 Hz since the excitation force increases with the square of the RPM. Figures 50 and 51 confirm the vibration on the 26-ft and 36-ft flats at the frequency of the primary structural resonance at about 10 Hz. This is reduced from 10.8 to 10 Hz by the flexibility of the supports. Finally, Figures 52 and 53 also show the primary vertical structural resonance of about 10 Hz and the motion on the bridge, which is less than on the 26- and 36-ft flats.

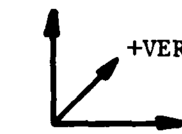
On a subsequent ship, a shaker was mounted on a thrust-bearing housing roughly bolted to its footing. The shafting was not connected. The ship was incomplete, but had most of its machinery in place. It was located in the fitting-out dock with little water under the keel. The model patterns along the bottom $\mathbb{C}_L$ at a number of frequencies are shown in Figures 54 and 55. In Figure 56 the experimentally determined responses on the bridge (at Frame 164) and on the bottom at Frame 163 are shown. The tests were run in the evening at the fitting-out dock under pressure to complete the ship. It would have been helpful in interpreting the data if the shaker had a circuit for providing a phase mark and if a few more readings had been taken.

It appears that the fundamental 10.6 Hz superstructure resonance is moved down to about 10 Hz by the larger water inertia of the shallow drafts. Since it is not strongly coupled with the shafting, its omission is not strongly felt. On the other hand, the resonance which appears at about 19.5 Hz probably corresponds in the bottom and the superstructure to the 12.4 Hz longitudinal shaft vibration mode without the shaft weight present.

2.3.26. Compare Measured Vibrations with Specifications. Although the measured vibrations were compared to the ISO whole body vibration standards, this comparison is not important to this evaluation of the computation procedures.

3. Procedure as Applied to a Large RO/RO Ship Design

A portion of the procedure outlined in this report was applied to the analysis of the vibration of a triple screw roll-on/roll-off trailer ship designed by Sun Shipbuilding and Drydock Company. The studies were made by Littleton Research and Engineering Corp., working closely with Mr. Hector McVey, project manager, and Charles Lofft. The program was an outstanding example of a good mutual interaction between ship vibration analysis and ship design. The ship's dimensions are 945-ft overall length, 875-ft length between perpendiculars, 105.5-ft beam, 76.25-ft depth, and 30-ft design draft. It is driven by a center and two wing propellers, each absorbing 50,000 rated horsepower at 155 RPM. Construction of the ship has not been initiated, and so comparisons between predictions and measurements are not possible. The correlation between the sequence of studies recommended earlier in this report and those applied to the vibration studies on this ship will be recognized by the brief summary of the reports generated in the course of the work. A listing of these reports is contained in Appendix G.



+ VECTOR PHASE 0° $\pm$ 90°

- VECTOR PHASE 180° $\pm$ 90°

AMPLITUDE IN MILS (0.001 INCH)

<u>26'-0" FLAT</u>

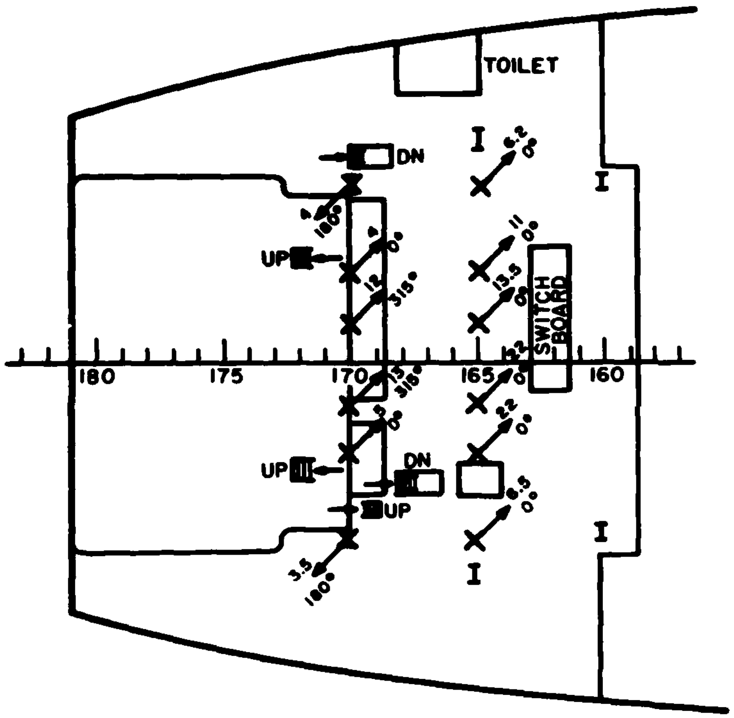
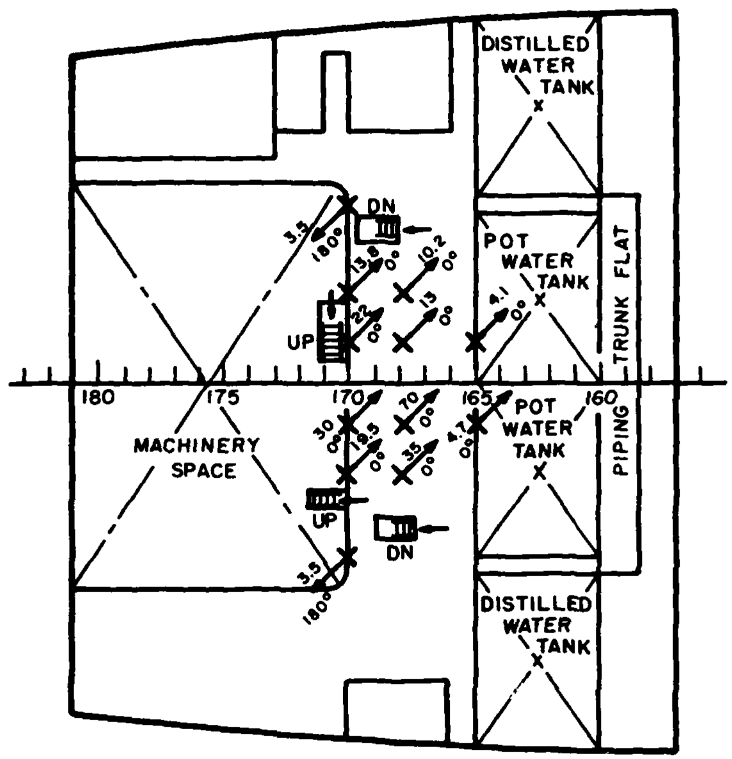
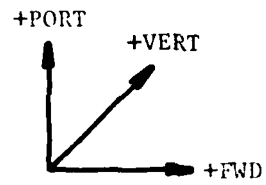


FIGURE 50. VIBRATION ON 26-FT FLAT AT ABOUT 100 RPM





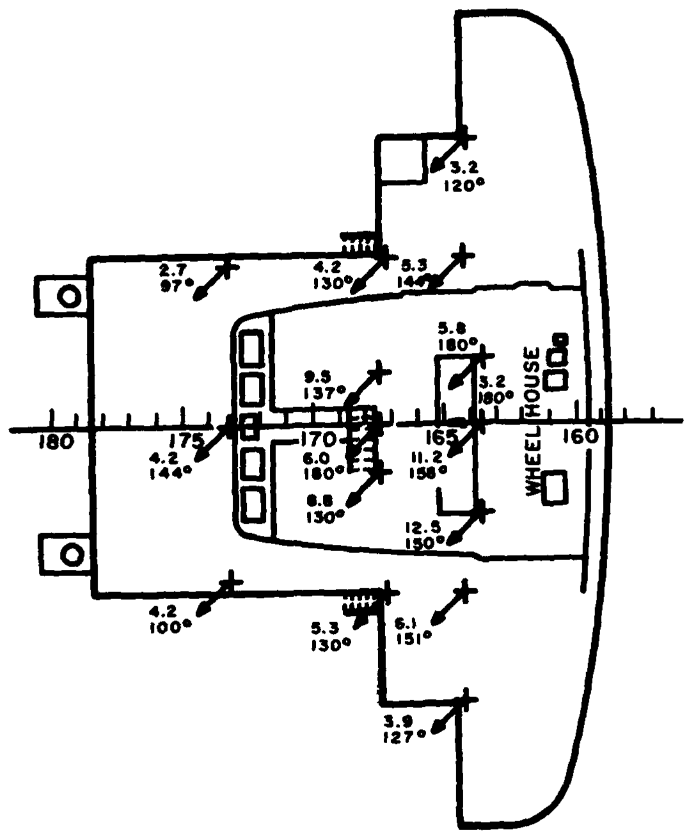
+ VECTOR PHASE 0° ± 90°

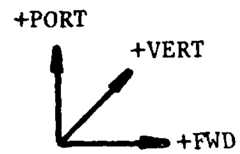
- VECTOR PHASE 180° ± 90°

AMPLITUDE IN MILS (0.001 INCH)

36'-0" FLAT

FIGURE 51. VIBRATION ON 36-FT FLAT AT ABOUT 100 RPM





+ VECTOR PHASE 0° ± 90°

- VECTOR PHASE 180° ± 90°

AMPLITUDE IN MILS (0.001 INCH)

FIGURE 52. VIBRATION OF BRIDGE DECK AT ABOUT 98.6 RPM

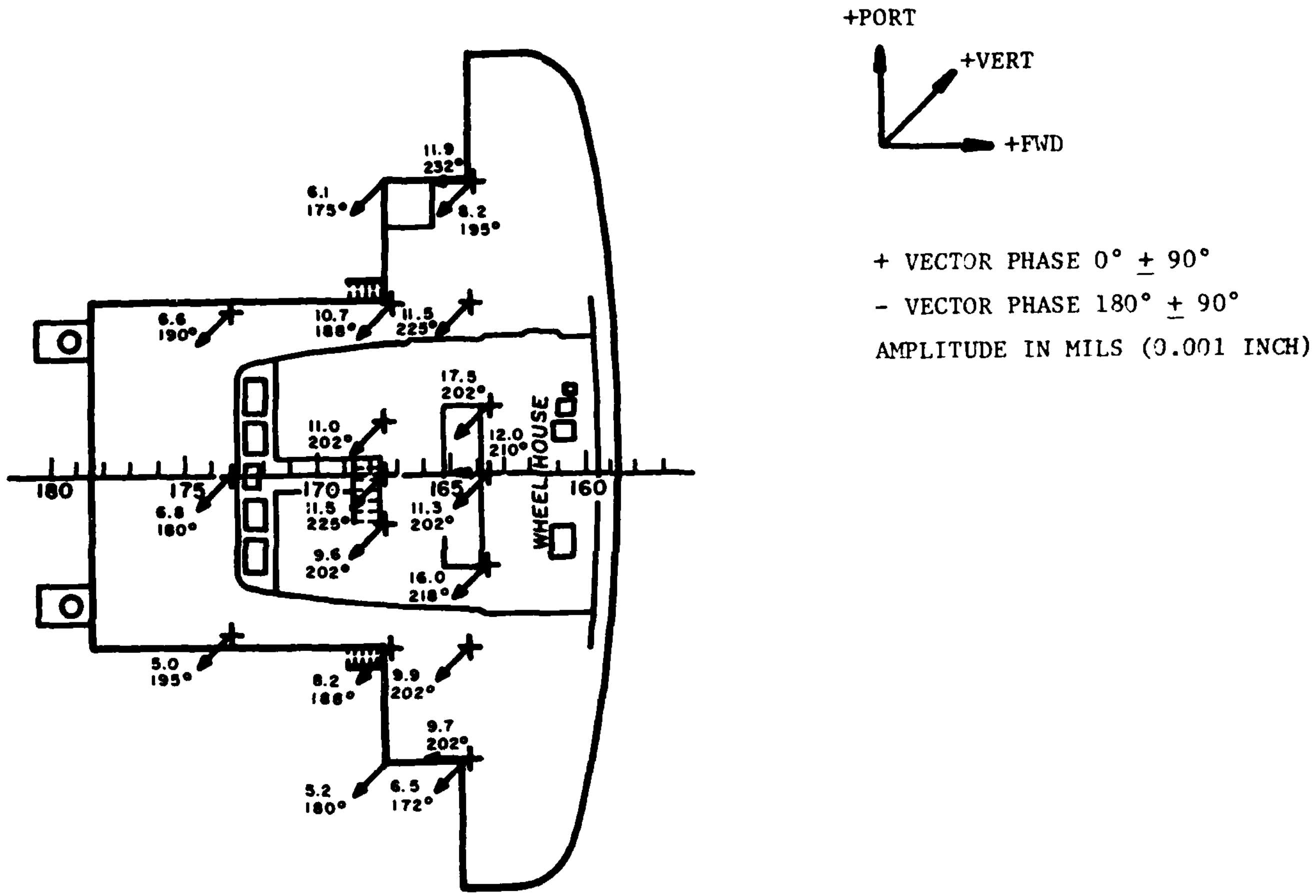


FIGURE 53. VIBRATION OF BRIDGE DECK AT ABOUT 102 RPM

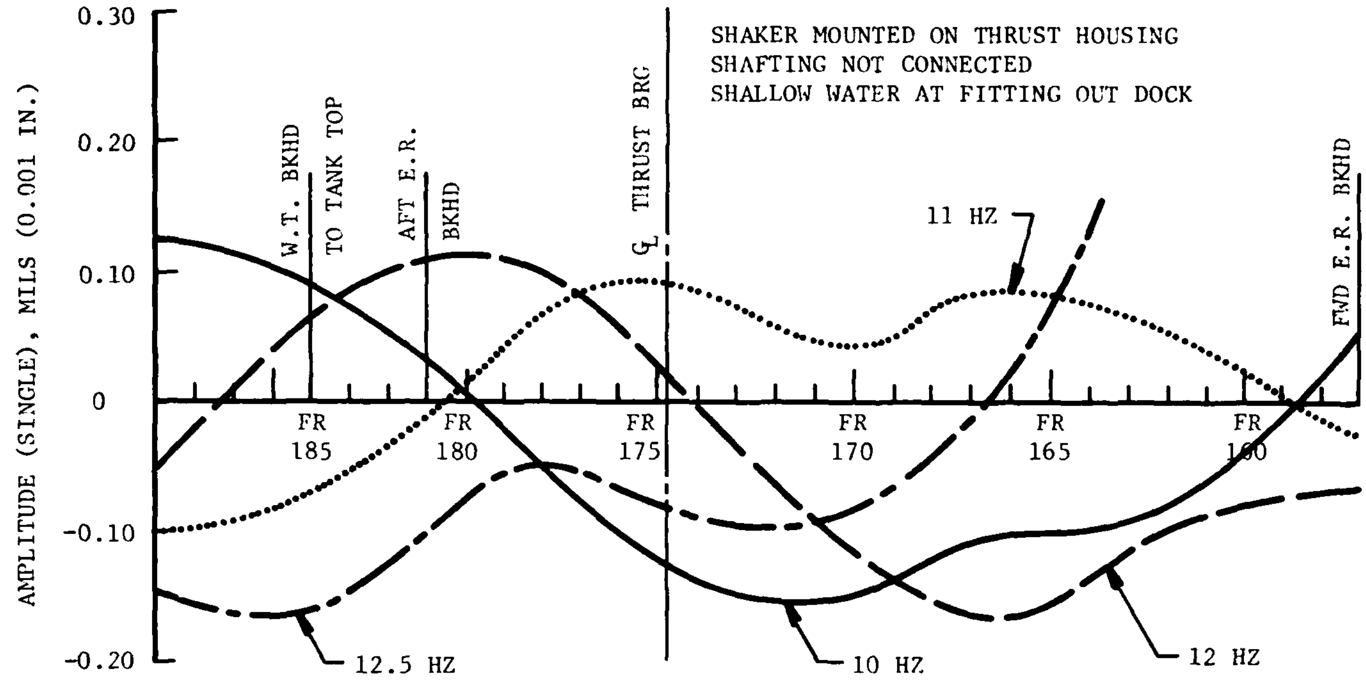


FIGURE 54. EXPERIMENTAL DEFLECTION PATTERNS ON TANK TOP AT ℄ AT 10, 11, 12, AND 12.5 HERTZ

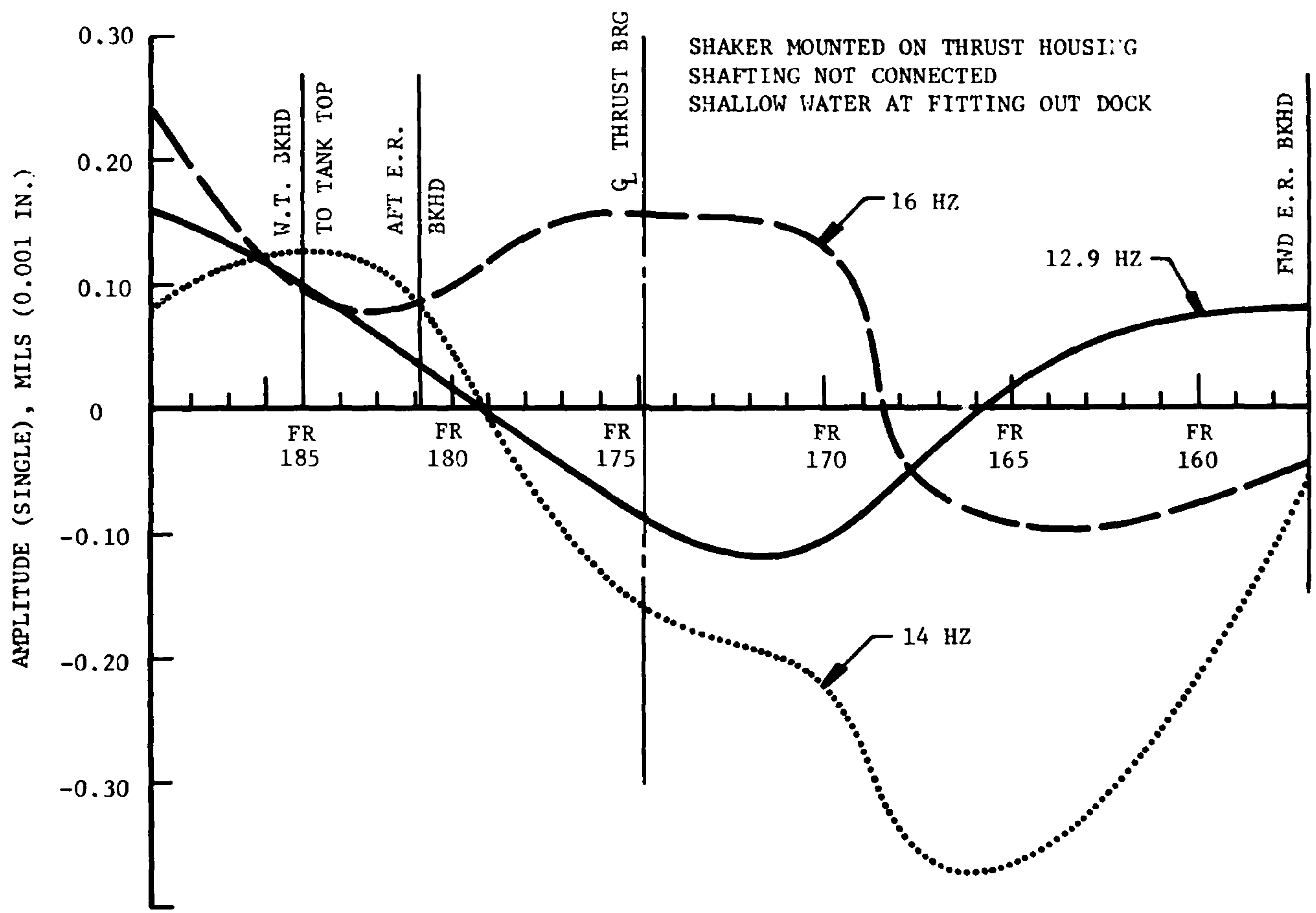


FIGURE 55. EXPERIMENTAL DEFLECTION PATTERNS ON TANK TOP AT ℄ AT 12.9, 14, AND 16 HERTZ

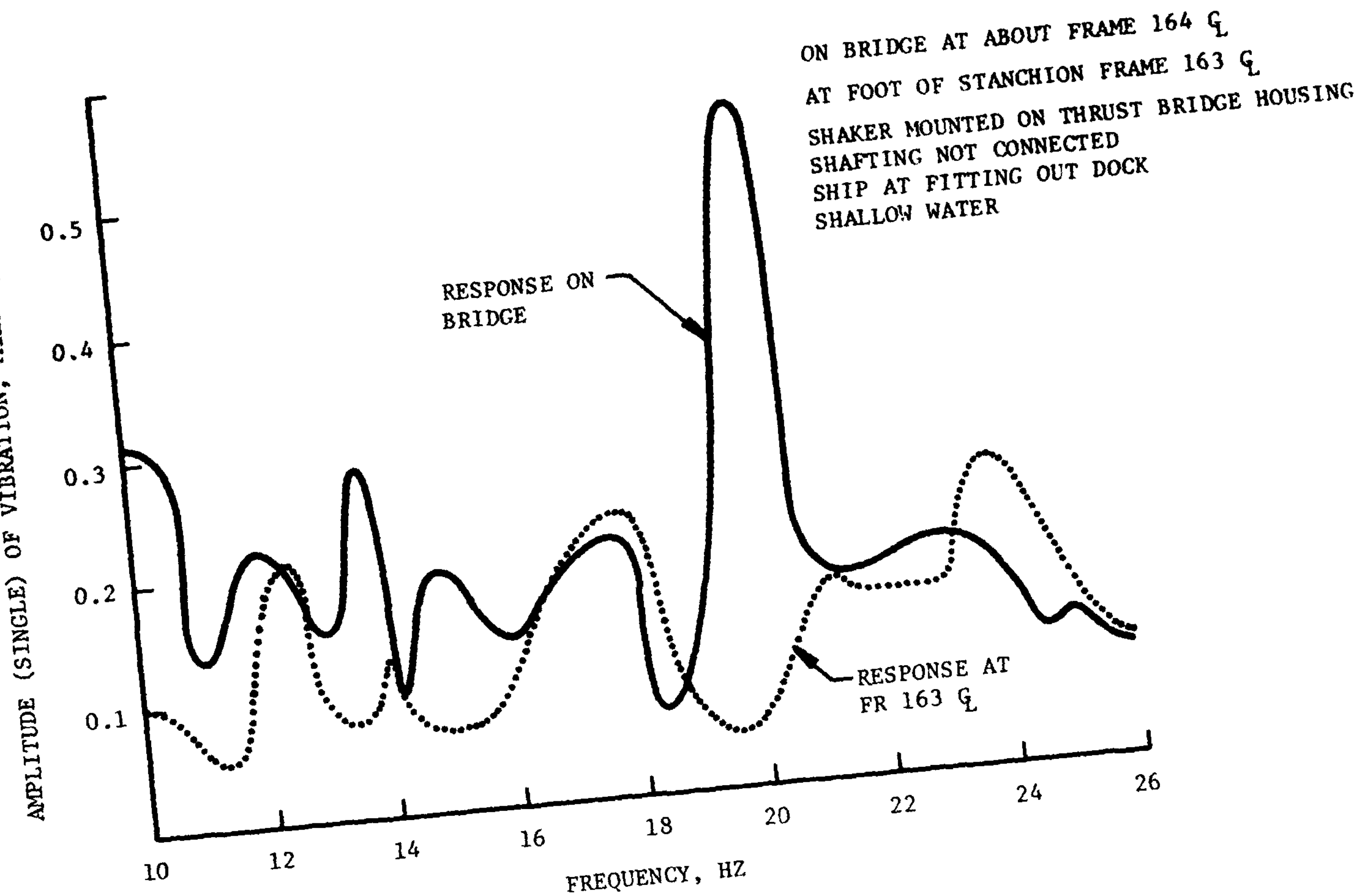


FIGURE 56. RESPONSE TO SHAKER EXCITATION

4. Discuss:on of Example Problem

In this chapter it has been demonstrated that, with regard to the design of the Litton Industries, Ingalls Shipbuilding Division United and Containerized Cargo Ship built for Farrell Lines, application of the finite analysis computer techniques now available would have forewarned of the following vibration difficulties experienced on the ship.

a. Longitudinal vibration of the propeller-shafting system. The analysis of the machinery space-superstructure subsystem indicated a natural frequency of 12.4 Hz. The frequency could be expected to fall when the subsystem is incorporated with the remaining ship structure and is too close to the maximum operating speed. The difficulty occurs because of coupling between the engine room double bottom modes and the vibration of the shafting system against the thrust bearing and foundation and would probably not be discovered by a static analysis.

b. The fundamental frequency of the machinery space and superstructure substructure in vertical vibration is 10.6 Hz. This frequency can also be expected to decrease when incorporated with the remaining hull structure. It coincides almost exactly with the blade frequency at full power and explains the heavy vibration on the 26- and 36-ft flats in the engine room, the vibration in the wheelhouse, and that in the passenger lounge.

c. The general vertical vibration levels in the ship are accentuated on the 26-ft and 36-ft machinery space flats. This was observed on trials. Since the main machinery control console is located on the 26-ft flat, this is a particularly bad location for vibration.

Time and money limitations did not permit the substructure predictions to be refined and quantified by being incorporated into a structure model of the complete ship.

A listing of the reports presented to Messrs. McVey and Lofft of Sun Shipbuilding and Dry Dock Company in the course of the design development of a large RO/RO ship demonstrates how a portion of the design procedure incorporating vibration analysis presented in this study was applied to a high-powered ship in which vibrations might be critical. Unfortunately, construction of this ship has not been initiated, so the accuracy of the procedures cannot be assessed for this case.

Although the analysis of the machinery space and superstructure on the Farrell Ship incorporates the best present techniques for predicting vibrations, its proposed combination with the remainder of the hull does not utilize the best presently available representations. It would be desirable to complete the study with continuing emphasis upon making the prediction as accurate as practically possible. All of the Farrell ships built to this design have been increased in size, and so it is not possible to make additional vibration measurements applicable to the initial studies.

Detailed calculations compared with detailed measurements under operating conditions and with shakers, comparable to the program carried on by McGoldrick at DTMB in the late 1940's and early 1950's, would appear desirable for developing confidence in vibration-prediction processes.

VI. CONCLUSIONS AND RECOMMENDATIONS

1. Conclusions

This report has presented a recommended design procedure for the minimization of propeller-induced vibrations in hull structural elements. The procedure is general in that it is not dependent upon the ship's size, type, or service conditions, and is intended to mesh with the overall ship's design process. The recommended procedure begins when the vibration specifications are defined and continues until after the vibration levels are measured during the ship's sea trials.

As presented, this recommended procedure is quite comprehensive and would be expensive to apply in its entirety. Therefore, the designer may choose to combine or eliminate some of the steps for the sake of economy. Such simplification can only be employed with great experience, and will be accompanied by increased risk of vibration problems occurring in the final design. A trade-off risk versus savings naturally results. Each designer must decide what part of this trade-off he is willing to accept.

Many conclusions pertaining to specialized parts of the design processes are contained in this report. Rather than reiterate them at this time, it is felt that the following list of more general conclusions regarding the entire procedure would be more beneficial to the reader:

- The technical state-of-the-art appears to be sufficiently advanced so that ships <u>can</u> be designed to have acceptable vibration levels.

- Present-day ships are complex mechanical systems whose successful design depends on extensive use of current analytical and experimental analysis methods.

- It is highly unlikely that "cookbook" type of design procedures can be developed and used successfully by the general maritime industry.

- The design process includes many specialized areas and requires the interaction of many individuals. It should be approached from a systems point-of-view, with equal emphasis given to propeller excitation and structural response.

- Any rational design procedure must, as a minimum, contain the following:

  - A set of design specifications

  - A method of predicting or measuring propeller-induced forces and pressures

  - A method of computing structural response

- A procedure for measuring actual vibration levels during sea trials.

2. <u>Recommendations</u>

Although the set of general conclusions stated that ships can be currently designed to have acceptable vibration levels, there are certain areas which require additional research if the recommended procedures are to be improved. They are:

- A study should be initiated to establish the relationships between the ship stern form to the wake and its variation, and the wake and wake variations to the propeller forces, hull pressure forces, and propeller cavitation.

- A study should be made to more fully understand the effects of a working propeller on the nominal wake distribution.

- Efforts should be increased to refine analytical methods to predict hull pressures caused by cavitation.

- A study should be initiated to establish guidelines for the acceptability of propeller-generated forces and pressures based upon gross ship properties.

- Additional work is needed in the area of propeller-rudder interaction. This includes improved prediction techniques for both the propeller-generated forces and the response of the rudder.

- It would be desirable to apply the procedures presented herein to a ship that has been built and is operating.

The designer of a ship should use discretion in the application of this procedure. This discretion should involve a review of each item of the process as applied to the ship. For a ship of low power, where vibration is rarely a problem unless there is a serious neglect in overlooking important factors that generate vibration, this review should give consideration to propeller clearance and the flow of water to the propeller, adequate support for the thrust bearing and reasonable support for outboard shafting. In ships that are similar to vibration-free ships already in operation, the items in the procedure should be checked between the ships.

In ships of moderate power, say less than 25,000 shaft horsepower, and of a design generally similar to other vibration-free ships, it may be adequate to check propeller clearance, to ensure that the ship has good lines for free flow to the upper portion of the propeller, that the thrust bearing appears to be well supported, and that the superstructure has good structural continuity.

With powers rising above 30,000 shaft horsepower and with the open structure characteristics of some of the recent roll-on/roll-off ships, it is probably desirable to run through the complete program. For balance between the different elements of the procedure, it is probably equally important for a vibration-free ship to have good structural studies as good hydrodynamic studies. If model studies include measurements of the pressures on the hull from cavitating propellers, then an equivalent degree of structural vibration analysis in the form of a study of the dynamic response of the machinery space and of the superstructure will be warranted.

The recommendations by the British Ship Research Association of checking the vibration response of a ship under excitation by a shaker at the stage where the structure is completed, but before the joiner work is installed, would appear to have some value for catching local resonances while it is inexpensive to correct them.

REFERENCES

1. Burnside, O. H., Kana, D. D., and Reed, F. E., "Bibliography for the Study of Propeller-Induced Vibration in Hull Structural Elements," SSC-281, Final Technical Report on Project SR-240, Department of Transportation, U. S. Coast Guard Contract No. DOT-CG-61907-A, July 1978.

2. Hagen, A., and Hammer, N. O., "Shipboard Noise and Vibration from a Habitability Viewpoint," Marine Technology, Vol. 6, No. 1, January 1969.

3. Reed, F. E., "Acceptable Levels of Vibration on Ships," Marine Technology, Vol. 10, No. 2, April 1973.

4. ISO Standard 2631, "Guide for the Evaluation of Human Exposure to Whole-Body Vibration," International Organization for Standardization, 1974.

5. The Society of Naval Architects and Marine Engineers, "Acceptable Vibration of Marine Steam and Heavy-Duty Gas-Turbine Main and Auxiliary Machinery Plants," Technical and Research Code C-5, September 1976.

6. The Society of Naval Architects and Marine Engineers, "Code for Shipboard Hull Vibration Measurements," Technical and Research Code C-1, January 1975.

7. The Society of Naval Architects and Marine Engineers, "Code for Local Shipboard Structures and Machinery Vibration Measurements," Technical and Research Code C-4, December 1976.

8. Van Oossanen, P., and Oosterveld, M. W. C., "Application of Theory in the Optimization of Single Screw Container Ship," presented at the International Symposium on Practical Design in Shipbuilding (PRADS), Tokyo, Japan, October 1977.

9. Hoekstra, M., "Prediction of Full Scale Wake Characteristics Based on Model Wake Surveys," International Shipbuilding Progress, Vol. 22, No. 250, June 1975.

10. Odabasi, A. Y., and Fitzsimmons, P. A., "Alternative Methods for Wake Quality Assessment," International Shipbuilding Progress, Vol. 25, No. 282, February 1978.

11. Van Gent, W., and van Oossanen, P., "Influence of Wake and Propeller Loading and Cavitation," Second LIPS Propeller Symposium, Drunen, Holland, May 1973.

12. The Society of Naval Architects and Marine Engineers, "Longitudinal Stiffness of Main Thrust Bearing Foundations," Technical and Research Bulletin R-15, September 1972.

13. Kane, J. R., and McGoldrick, R. T., "Longitudinal Vibration of Marine Propulsion-Shafting System," Transactions, SNAME, Vol. 57, pp 195-252, 1949.

14. Mott, I. K., and Fleeting, R., "Design Aspects of Marine Propulsion Shafting Systems," Transactions, Institution of Marine Engineers, June 1967.

15. Lewis, F. M., and Auslaender, J., "Virtual Inertia of Propellers, Journal of Ship Research, Vol. 3, No. 4, March 1960.

16. Reed, F. E., and Burnside, O. H., "Computer Techniques for Use in Ship Hull Vibration Analysis and Design," Paper K, SSC and SNAME Ship Vibration Symposium '78, Arlington, Virginia, October 1978.

17. Saunders, H. E., Hydrodynamics in Ship Design, SNAME, New York, 1957.

18. Kerwin, J. E., "Computer Techniques for Propeller Blade Section Design," International Shipbuilding Progress, Vol. 20, No. 227, July 1973.

19. Van Gunsteren, L. A., and Pronk, C., "Propeller Design Concepts," International Shipbuilding Progress, Vol. 20, No. 227, July 1973.

20. Cox, C. G., and Morgan, W. B., "The Use of Theory in Propeller Design," Marine Technology, Vol. 9, No. 4, October 1972.

21. Cummings, R. A., Morgan, W. B., and Boswell, R. J., "Highly Skewed Propellers," Transactions, SNAME, Vol. 80, pp 98-135, 1972.

22. Boswell, R. J., and Cox, C. G., "Design and Model Evaluation of a Highly Skewed Propeller for a Cargo Ship," Marine Technology, Vol. 11, No. 1, January 1974.

23. Lewis, F. M., "Propeller Vibration," Transactions, SNAME, Vols. 43 and 44, 1935-1936.

24. Burrill, L. C., "Calculation of Marine Propeller Characteristics," Northeast Coast Institute of Engineers and Shipbuilders, Vol. 60, 1943-1944.

25. Hinterthon, W. B., "Propeller-Excited Vibratory Forces for the Tanker *SS ESSO Gettysburg* Calculated from Wake Surveys," Naval Ship Research and Development Report 2870, August 1968.

26. Reed, F. E., and Bradshaw, R. T., "Ship Hull Vibrations II. Distribution of Exciting Forces Generated by Propellers," CONESCO Report to Bureau of Ships, F-101-2, Index No. NS712-100ST2, June 1960.

27. Tsakonas, S., and Jacobs, W. R., "Documentation of a Computer Program for the Pressure Distribution Forces and Moments on Ship Propellers in Hull Wakes," Stevens Institute of Technology, Davidson Laboratory Report SIT-DL-76-1863.

28. Brown, N. A., "Periodic Propeller Forces in Non-Uniform Flow," Massachusetts Institute of Technology, Department of Naval Architecture and Marine Engineering Report 64-7, 1964.

29. Frydelund, O., and Kerwin, J. E., "The Development of Numerical Methods for the Computation of Unsteady Propeller Forces," Symposium on Hydrodynamics of Ship and Offshore Propulsion Systems, Det norske Veritas, Oslo, Norway, March 1977.

30. Murray, M. T., and Tubby, J. E., "Blade-Rate Force Fluctuations of a Propeller in Non-Uniform Flow," Admiralty Research Laboratory, ARL/M/P33A, June 1973.

31. McGoldrick, R. T., "Ship Vibration," David Taylor Model Basin Report 1451, December 1960.

32. Breslin, J. P., "Techniques for Estimating Vibratory Forces Generated by Propellers," SNAME Technical and Research Bulletin No. 1-34, January 1975.

33. Van Oossanen, P., "Theoretical Prediction of Cavitation on Propellers," Marine Technology, Vol. 14, No. 4, October 1977.

34. Huse, E., "The Magnitude and Distribution of Propeller-Induced Surface Forces on Single-Screw Ship Model," Publication No. 100, Norwegian Ship Model Experiment Tank, December 1968.

35. Frivold, H., "Solid Boundary Factors for the Afterbody of an LNG Carrier," Norwegian Maritime Research, Vol. 4, No. 1, 1976.

36. Vorus, W. S., "A Method for Analyzing the Propeller-Induced Vibratory Forces Acting on the Surface of a Ship Stern," Transactions, SNAME, Vol. 82, 1974.

37. Yildiz, M., and Mawardi, O. K., "On the Diffraction of Multiple Fields by a Semi-Infinite Rigid Wedge," Journal of the Acoustical Society of America, Vol. 32, No. 12, December 1960.

38. Yildiz, M., and Mawardi, O. K., "Diffraction of a Dipole Field by a Rigid Cone (Abstract)," Journal of the Acoustical Society of America, Vol. 32, No. 11, November 1960.

39. Johnsson, C. A., "Pressure Fluctuations Around a Marine Propeller--Results of Calculations and Comparison with Experiment," Swedish State Shipbuilding Experimental Tank Report No. 69, 1971.

40. Breslin, J. P., and Eng, K. S., "A Method of Computing Propeller-Induced Vibratory Forces on Ships," presented at First Conference on Ship Vibrations, Stevens Institute of Technology, January 1965.

41. Tsakonas, S., Jacobs, W. R., and Ali, M. R., "Documentation for the Computer Program for the Pressure Field Generated by a Propeller in a Variable Inflow," Stevens Institute of Technology, Davidson Laboratory Report 1910, May 1977.

42. Vorus, W. S., "Calculation of Propeller-Induced Vibratory Hull Forces, Force Distributions, and Pressures; Free-Surface Effects," Journal of Ship Research, Vol. 20, No. 2, June 1976.

43. Johnsson, C. A., "Correlation of Predictions and Full Scale Observations of Propeller Cavitation," International Shipbuilding Progress, Vol. 20, No. 226, June 1973.

44. Lewis, F. M., and Kerwin, J. E., "Vibratory Forces on a Simulated Hull Surface Produced by Transient Propeller Cavitation," Journal of Ship Research, Vol. 22, No. 2, June 1978.

45. Takahashi, H., and Ueda, T., "An Experimental Investigation into the Effect of Cavitation on Fluctuating Pressures Around a Marine Propeller," 12th International Towing Tank Conference, Rome, 1969.

46. Van Manen, J. D., "The Effect of Cavitation on the Interaction Between Propeller and Ship's Hull," International Shipbuilding Progress, Vol. 19, No. 209, January 1972.

47. Søntvedt, T., "Propeller Induced Excitation Forces," Det norske Publication No. 74, January 1971.

48. Van Oossanen, P., and van der Kooy, J., "Vibratory Hull Forces Induced by Cavitating Propellers," Proceedings, Royal Institution of Naval Architects, Vol. 115, 1973.

49. Johnsson, C. A., and Søntvedt, T., "Propeller Excitation and Response of 230,000 TDW Tankers," Swedish State Shipbuilding Experimental Tank Publication No. 70, 1972. Also presented at the Ninth Symposium on Naval Hydrodynamics, Paris, August 1972.

50. Johnsson, C. A., et al., "Vibration Excitation Forces from a Cavitating Propeller-Model and Full Scale Tests on a High Speed Container Ship," Swedish State Shipbuilding Experimental Tank Publication No. 78, 1976.

51. Noordzij, L., "Pressure Field Induced by a Cavitating Propeller," International Shipbuilding Progress, Vol. 23, No. 260, April 1976.

52. Lewis, F. M., "Propeller-Vibration Forces in Single-Screw Ships," Transactions, SNAME, Vol. 77, 1969.

53. Lewis, F. M., "Propeller Excited Hull and Rudder Force Measurements," Massachusetts Institute of Technology, Department of Ocean Engineering Report 73-10, April 1973.

54. Huse, E., "Cavitation-Induced Hull Pressures, Some Recent Developments of Model Testing Techniques," Norwegian Maritime Research, Vol. 3, No. 2, 1975.

55. Oosterveld, M. W. C., and van Oossanen, P., "Recent Results of Tests in the NSMB Depressurized Towing Tank," First Ship Technology and Research (STAR) Symposium, Washington, D. C., August 1975.

56. Norita, H., Kuritake, Y., and Yagi, H., "Application and Development of a Large Ducted Propeller for the 288,000-dwt Tanker *MS Thorsage*," Transactions, SNAME, Vol. 82, 1972.

57. Okamoto, H., et al., "Cavitation Study of Ducted Propellers on Large Ships," Transactions, SNAME, Vol. 83, 1975.

58. Dyne, G., and Hoekstra, M., "Propulsion Cavitation and Propeller-Induced Pressure Fluctuations of a Tanker (Comparative Tests in SSPA Cavitation Tunnel No. 2 and NSMB Depressurized Towing Tank)," SNAME, Spring Meeting, Philadelphia, Pennsylvania, June 1976.

59. The National Shipbuilding Research Program--Research on Computer Applications to Shipbuilding VII, Catalog of Program Abstracts, U. S. Department of Commerce, Maritime Administration, in cooperation with Avondale Shipyards, Inc., May 1975.

60. Hananel, A. S., et al., "Static and Dynamic Analyses of the LSES Hull Structure," Journal of Ship Research, Vol. 22, No. 2, June 1978.

61. Larsen, O. C., "Some Considerations on Shaft Alignment of Marine Shafting," Norwegian Maritime Research, Vol. 4, No. 2, 1976.

62. Reed, F. E., "The Design of Ships to Avoid Propeller-Excited Vibrations," Transactions, SNAME, Vol. 79, 1971.

63. Bureau Veritas Publications, Machinery Hull Interaction--Vibrations, Paris, France, 1975.

64. Volcy, G. C., Garnier, H., and Masson, J. C., "Deformability of the Hull Steelwork and Deformations of the Engine Room of Large Tankers," published in Machinery Hull Interaction--Vibrations, Bureau Veritas, Paris, 1975.

65. Restad, K., "Investigation on Free and Forced Vibrations of an LNG Tanker with Overlapping Propeller Arrangement," published in Machinery Hull Interaction--Vibrations, Bureau Veritas, Paris, 1975.

66. Volcy, G. C., Garnier, H., and Masson, J. C., "Chain of Static and Vibratory Calculations of Propulsive Plants and Engine Rooms of Ships," published in Machinery Hull Interaction--Vibrations, Bureau Veritas, Paris, 1975.

67. Senjanovic, I., and Skaar, K. T., "Phenomena of Ship Vibration," Det norske Veritas Publication No. 88, May 1975.

68. Bureau Veritas, "Recommendations Designed to Limit the Effects of Vibrations on Board Ships," Guidance Note, N.I. 138 BM.3E, May 1971.

69. Volcy, G. C., "Forced Vibration of the Hull and Rational Alignment of the Propeller Shaft," published in Machinery Hull Interaction--Vibrations, Bureau Veritas, Paris, 1975.

70. Bourceau, G., and Volcy, G. C., "Forced Vibration Resonators and Free Vibration of the Hull," published in Machinery Hull Interactions--Vibrations, Bureau Veritas, Paris, 1975.

71. Hylarides, S., "Transverse Vibrations of Ship's Propulsion System, Part I, Theoretical Analysis," International Shipbuilding Progress, Vol. 21, No. 252, August 1975.

72. Vassilapoulos, L., and Bradshaw, R., "Coupled Transverse Shaft Vibrations of Modern Ships," International Shipbuilding Progress, Vol. 21, No. 239, July 1974.

73. Golden, M. E., and Henderson, F. M., "An Updated Guide to the Use of General Bending Response Program (GBRP)," Computation and Mathematics Department Research and Development Report 4601, Naval Ship Research and Development Center, Bethesda, Maryland, April 1975.

74. Volcy, G. C., and Osouf, J., "Vibratory Behaviour of Elastically Supported Tail Shafts," published in Machinery Hull Interaction--Vibrations, Bureau Veritas, Paris, 1975.

75. Hansen, H. R., and Skaar, K. T., "Hull and Superstructure Vibrations Design Calculation by Finite Elements," Det norske Veritas Publication 86, January 1975.

76. Johannessen, H., Skaar, K. T., and Sunogeli, H., "Dynamic Response of Hull and Superstructure," presented at the International Symposium on Practical Design in Shipbuilding (PRADS), The Society of Naval Architects of Japan, Tokyo, October 1977.

77. Carlsen, C. A., "A Parametric Study on Global Hull and Superstructure Vibration Analysis by Means of the Finite-Element Method," Paper Issued for Written Discussion, The Royal Institute of Naval Architects, 1977.

78. Meyerhoff, W. K., "Added Masses of Thin Rectangular Plates Calculated from Potential Theory," Journal of Ship Research, Volume 14, No. 2, June 1970.

79. Lindholm, U. S., Kana, D. D., Chu, W. H., and Abramson, H. N., "Elastic Vibration Characteristics of Cantilever Plates in Water," Journal of Ship Research, Vol. 9, No. 1, June 1965.

80. Marcus, M. S., "A Finite-Element Method Applied to the Vibration of Submerged Plates," Journal of Ship Research, Vol. 22, No. 2, June 1978.

81. Tsakonas, S., Jacobs, W. R., and Ali, M. R., "Application of the Unsteady-Lifting-Surface Theory to the Study of Propeller-Rudder Interaction," Journal of Ship Research, Vol. 14, No. 3, September 1970.

82. Tsakonas, S., Jacobs, W. R., and Ali, M. R., "Propeller-Rudder Interaction Due to Loading and Thickness Effects," Journal of Ship Research, Vol. 19, No. 2, June 1975.

83. Leissa, A. W., "Vibration of Plates," National Aeronautics and Space Administration Publication NASA SP-160, Washington, D. C., 1969.

84. Robinson, D. C., "Ship Vibration Beam Theory: Part I - A Review of the Theory," Report AVL-76-962, Acoustics and Vibration Laboratory, Vibration Division, David Taylor Model Basin, Washington, D. C., August 1965.

85. Jensen, J. J., and Madsen, N. F., "A Review of Ship Hull Vibration. Part I: Mathematical Models; Part II: Modeling Physical Phenomena; Part III: Methods of Solution; Part IV: Comparison of Beam Models," The Shock and Vibration Digest, Vol. 9, Nos. 4-7, April-July 1977.

86. Leibowitz, R. C., and Harder, R. L., "Mechanized Computation of Ship Parameters," David Taylor Model Basin Report 1841, June 1965.

87. Volcy, G. C., Garnier, H., and Masson, J. C., "An Analysis of the Free and Forced Vibrations of Cargo Tank Structure by Finite-Element Technique," HANSA, Heft 9, Hamburg, 1975.

88. Skaar, K. T., "On the Finite-Element Stress Analysis of Oil Tanker Structures," Norwegian Maritime Research, Vol. 2, No. 2, 1974.

89. Kamel, H. A., and Liu, D., "Structural Dynamic Problems in Ships and Other Marine Structures," contained in Finite-Element Applications in Vibration Problems, edited by M. M. Kamel and J. A. Wolf, Jr., The American Society of Mechanical Engineers, New York, September, 1977.

90. Egeland, O., and Araldsen, P. O., "SESAM-69 - A General Purpose Finite Element Program," Computers and Structures, Vol. 4, No. 1, January 1974.

91. Kumai, T., "Damping Factors in the Higher Modes of Ship Vibration," Reports of Research Institute for Applied Mechanics, Vol. VI, No. 21, Kyushu University, Fukuoka, Japan, 1958.

92. Woolam, W. E., "Research on Ship-Hull Damping Coefficients for Low-Frequency Vertical Flexural Modes of Vibration," Acoustics and Vibration Laboratory Research and Development Report 2323, Naval Ship Research and Development Center, May 1967.

93. Volcy, G. C., and Nakayama, M., "Studies Leading to Vibration and Noise Free Ships," Bulletin Technique Du Bureau Veritas (Special English Issue), Bureau Veritas, Paris, June 1977.

94. Hart, H. H., "Hull Vibrations of the Cargo Liner *Koudekerk*," International Shipbuilding Progress, Vol. 18, No. 206, October 1971.

95. Knopfle, W. H., "Vibration Survey Techniques," Marine Technology, Vol. 8, No. 1, January 1971.

96. Kuhii, K., "Ship Vibration Measurement and Analysis by Data Acquisition and Processing System," IHI Engineering Review, Special Issue '71 Ships, Ishikawajima-Harima Heavy Industries Co., Ltd., Tokyo, Japan, July 1971.

97. Solumsmoen, O. H., "Ship Vibration - Experience from Service Measurements," Det norske Veritas Publication No. 96, January 1977.

98. Okamoto, H., "Reduction of Hull Vibration by Proper Selection of Propeller Type and Stern Form," Bulletin of the Marine Engineers Society of Japan, Vol. 5, No. 2, June 1977.

99. Huse, E., "Effect of Afterbody Forms and Afterbody Fins on the Wake Distribution of Single Screw Ships," Norges Skipsforskning-institut, Report R-31.74, Trondheim, Norway, 1974.

100. Huse, E., "Über propellererregte Vibrationen auf Schiffen," Hansa, No. 12, 1970.

101. Huse, E., "Trykkimpulser fra Kaviterende propell," Nordiskt Skeppstekniskt Möte (NSTM), 1971. (In Norwegian)

102. Lindgren, H., and Johnssen, C. A., "On the Influence of Cavitation on Propeller Excited Vibratory Forces and Some Means of Reducing its Effect," presented at the International Symposium on Practical Design in Shipbuilding (PRADS), Tokyo, Japan, October 1977.

103. Johnsson, C. A., "Stern Vibrations of High-Powered Ships," Final Report No. 1763-14, Swedish State Shipbuilding Tank, Göteborg, Sweden, August 1975.

104. Johnsson, C. A., "Hydrodynamically Generated Hull Vibrations--The Propeller as Excitation Source," Nordiskt Skeppstekniskt Möte (NSTM), Lyngby, Denmark, 1977. (In Swedish)

105. Sartor, T. J., and Giblon, R. P., "Farrell Lines '85' - Class Containerships," Marine Technology, Vol. 16, No. 1, January 1979.

106. Morse, P. M., Vibration and Sound, McGraw-Hill Book Co., Inc., 1948.

107. National Steel and Shipbuilding Company, "Highly-Skewed Propeller Research Program, San Clemente Ore/Bulk/Oil (OBO) Carriers," Final Report to Maritime Administration on Contract 2-36315, November 1974.

108. Littleton Research and Engineering Corp., "Measurement and Analysis of Harmonic Hull Pressures and Vibration Generated by Normal Propellers on the Seabridge Class RO/RO Ships," Report C-234-2 to American Export Lines, November 1974.

APPENDIX A

COMPUTER PROGRAMS FOR COMPUTING PROPELLER FORCES AND MOMENTS

A-1. Propeller Mean and Vibratory Forces Program

(Program developed by Davidson Laboratory, Stevens Institute of Technology, under U.S. Navy contracts; widely used.)

Output Information

1. Steady and time-dependent blade loading distribution at multiples of shaft frequency.

2. Mean and blade-frequency force and moment components in coefficient form for:

   (a) Thrust/$\rho n^2 d^4$

   (b) Torque/$\rho n^2 d^5$

   (c) Transverse force/$\rho n^2 d^4$

   (d) Vertical force/$\rho n^2 d^4$

   (e) Transverse bending moment/$\rho n^2 d^5$

   (f) Vertical bending moment/$\rho n^2 d^5$

where

ρ = fluid density

n = propeller rpm

d = propeller diameter

3. Blade bending moments about the pitch line at various radial positions and for various orders of excitation.

4. Information for the study of cavitation inception.

5. Information for the study of blade stress analysis which is performed by utilizing the STARDYNE-CDC finite element computer program.

Input Information

1. The propeller blade geometry.

2. The Fourier components of the spatial variation of the axial and tangential components of the wake.

Basis for Calculation

The program uses unsteady lifting-surface theory and takes into consideration all the relevant propeller geometry and the spatial nonuniformity of the inflow field. The program is available through Davidson Laboratory for $6,000. See Reference 27.

A-2. Harmonic Forces and Moments Generated by a Propeller in Nonuniform Flow

(Program developed and used by Littleton Research and Engineering Corp.)

Output Information

1. Magnitude and phase of the three components of harmonic propeller force and the three components of harmonic propeller moment.

2. The steady vertical and horizontal forces and moments arising from first-order wake action (thrust offset).

Input Information

1. Propeller drawing. The propeller drawings should show the following information: propeller diameter, hub diameter, rake, number of blades and propeller material; the variation with radius of chord, skewback, and pitch; propeller sections at several radii showing the variation of thickness along the chord. For propellers designed in Europe, the variation with radius of the distance from the reference line to the leading edge, trailing edge, and point of maximum thickness is acceptable in place of the variation of chord and skewback.

2. Ship speed and corresponding shaft rpm.

3. Wake as measured in a model test. The results of a harmonic analysis of the measured wake are required. If the harmonic analysis results are not available, the measured inflow velocities specified at several points along the radius and at frequent points around the circumference are acceptable, and a harmonic analysis will be performed.

   If a measured wake is not available, it can be inferred from the available wakes of other ships.

Basis for Calculations

Propeller forces are determined by lifting line theory. This is much less complex than the Davidson Laboratory lifting surface theory, but is considered adequate in view of uncertainties in the wake and the wide variation in service wake due to ship motions and sea action. The main reason for continuing to use the lifting line theory calculation is that it is the basis for the predictions of hull pressure and hull forces (see Appendix B-1).

This is a proprietary program not developed for general distribution.

A-3. <u>Calculation of Steady and Harmonic Propeller Forces</u>

(Computer program used by the American Bureau of Shipping.)

<u>Output Information</u>

1. Mean and blade-frequency components of the three forces and three moments acting on the propeller.

2. Time-varying blade pressure distribution at each wake harmonic.

<u>Input Information</u>

1. Propeller blade geometry.

2. Fourier coefficients of the spatial variation of the axial and tangential components of wake.

<u>Basis of Calculation</u>

The program employs an extended version of unsteady lifting line theory as developed by Dr. Neal A. Brown at MIT [28]. The extension includes the effects of propeller skew, which were not treated in the original theory. The results of this program are used as partial input to the "Surface Force" program described in Appendix B-3.

A-4. <u>Harmonic Forces and Moments Generated by a Propeller in Nonuniform Flow</u>

(A computer program under development at Massachusetts Institute of Technology by Professor Justin E. Kerwin, Department of Ocean Engineering.)

The program represents the propeller blade by grid points distributed over the surface and the wake spatially defined (cylindrical coordinates) in three directions: longitudinal, tangential, and radial. A distribution of vorticity is assumed over the surface, and by successive iterations is refined to be compatible with the boundary of the propeller surface and the laws of hydrodynamics, Kelvin's theorem, and the Kutta requirement for flow continuity at the trailing edge.

This discrete element approach appears to offer a number of advantages as a starting point for the computation of unsteady, partially cavitating flows:

(a) It is capable of yielding accurate predictions of mean loading, both at design and off-design conditions.

(b) Being a numerical procedure, blade geometry can be incorporated exactly so that propellers with large skew, rake,

and varying pitch distribution can be accommodated. This is considered essential, since it is through the variation of these parameters that optimum propeller designs can be evolved.

(c) Since the procedure includes all three components of induced velocity, there is no particular problem in including tangential and radial wake field components.

(d) Since no loading mode functions are employed, the modifications ultimately required to include the cavities would appear to be feasible. Source elements presently included to represent blade thickness can assume the further role of representing the cavity volume.

(e) A discrete element method lends itself naturally to a step-by-step domain solution, which is also essential for the subsequent inclusion of unsteady cavitation.

The procedures are still under development, but have been applied to specific cases with good results. See References 18 and 29.

A-5. Harmonic Forces and Moments Generated by a Propeller in Nonuniform Flow

(Computer program developed and used by the Admiralty Research Laboratory, Teddington, England.)

Output Information

1. The input data.

2. If wakes are given as velocity measurements, the harmonic values are printed (to the 71 harmonic). If given as Fourier components, these are listed.

3. The contribution to thrust, torque, vertical and horizontal forces, and moments from each specified radial section.

4. The integrated thrust, torque, horizontal and vertical forces, and moments for multiples of blade rate harmonics.

Input Information

1. Shaft speed.

2. Propeller geometry, including skew, chord length, blade pitch angle at specific radii.

3. Wake, either in Fourier Series, amplitude-plus-phase form, or as equally spaced measurements of wake at the radii where the propeller geometry information is given. Only axial or both axial and tangential wakes may be specified.

4. Calculations can be run for successive skew values.

5. Input radii may vary from 4 to 14.

6. As many as 20 skew configurations may be determined.

7. As many as 140 harmonics of the blade frequency forces may be calculated, but generally the number is limited to 10.

8. As many as 100 wake harmonics and 200 wake measurements per radius may be input.

Basis for Calculation

The calculation of the fluctuating forces on a propeller falls into three parts. The first part is the calculation of the variation of the inflow velocity to the blades; the next stage involves the calculation of the fluctuating lift-distribution on a section of blade associated with this fluctuating inflow; the final stage is the calculation of the propeller forces and moments. The calculation of the fluctuating lift is based on two-dimensional unsteady airfoil theory. It ignores blade-to-blade interaction and the variation with radius of the various significant parameters. These approximations would be unacceptable for predicting the steady lift, but are acceptable for the unsteady lift, probably overestimating the lift. See Reference 30.

APPENDIX B

COMPUTER PROGRAMS FOR COMPUTING HULL PRESSURES AND FORCES

B-1. Calculation of Harmonic Forces and Moments on the Hull Generated by Propeller Action

(Program developed and used by Littleton Research and Engineering Corp.)

Output Information

1. The harmonic hull surface pressure at blade-rate frequency generated by the loaded, noncavitating propeller in the region of the propeller (generally at grid points corresponding to underwater intersections of buttocks and frames within four diameters of the propeller).

2. By integration of the above, the blade frequency harmonic hull forces and moments acting on the hull due to noncavitating propeller action.

Input Information

1. The computed propeller lift distribution along the propeller blade (see Appendix A-2).

2. The geometry of the propeller.

3. The hull coordinates at the points of pressure determination.

Basis for Calculation

The free-field pressures (i.e., the pressures that would exist in open water if the hull were not present) are calculated at each hull grid point due to (1) the loading on the propeller blades (assumed to be concentrated at the forward quarter point of the blade chord), (2) the thickness of the propeller blade. The sum of these two pressures, in their proper phase, is multiplied by two to give the reported pressure on the hull surface. The pressure from a harmonically varying force having x, y, and z components involves the distance from the point to the location of the force. Substituting steady and harmonic forces and distances as a function of shaft angle yields values of the pressure. The resulting equations involve a series which under certain conditions converges slowly. Originally, only a few terms were developed. More recently, the general term has been developed, allowing sufficient terms to assure convergence. This results in pressures that correspond to measured values. The integration for blade thickness is similar. If the cavitation volume on the blade could be defined by a Fourier series, the same process could be applied. This has not yet been done.

This is a proprietary program not developed for general distribution. See Reference 26.

B-2. Calculation of Steady and Harmonic Pressure Fields Generated by a Noncavitating Propeller

(Program developed by Davidson Laboratory, Stevens Institute of Technology, under U.S. Navy contracts.)

Output Information

This program furnishes the steady and harmonic components of the pressure field generated by a noncavitating ship propeller operating in a spatially variable inflow.

Input Information

1. The propeller blade geometry.

2. The Fourier components of the spatial variation of the axial and tangential variation of the axial and tangential components of the wake.

3. The spatial location of the points where the pressures are desired.

4. The steady and time-dependent blade loading distribution at multiples of any shaft frequency as produced by the program described in Appendix A-1.

Basis for Calculation

This program is a continuation of the one described in Appendix A-1 and requires data generated in that program. It is available through Davidson Laboratory for $5,000. See Reference 41.

B-3. Calculation of Propeller-Induced Hull Surface Forces

(Program developed and used by Professor William S. Vorus (University of Michigan) and the American Bureau of Shipping.)

Output Information

This program computes all components of the hull force and moment at multiples of the propeller blade rates. (In general, the vertical force component is the only one desired.)

Input Information

1. Propeller geometry.

2. Wake distribution.

3. Stern lines and coordinates describing the sectional geometry of approximately the aft one-third of the ship.

4. Time-dependent geometry of propeller cavitation effects (optional).

5. Time-varying blade pressure distribution at each wake harmonic (output from program described in Appendix A-3).

Basis of Calculation

This program employs the method presented by Professor William S. Vorus in Reference 36 . The conventional procedure of evaluating the hull forces is to integrate the propeller-generated pressures over the hull surface. These pressures are due to diffraction of the propeller-induced water flow by the hull. The diffraction problem and hence the pressure integration difficulties are avoided in the analysis and computer program by utilizing a special application of Green's Theorem.

APPENDIX C

COMPUTER PROGRAMS FOR COMPUTING THE LONGITUDINAL RESPONSE OF THE PROPULSION SHAFTING

C-1. Longitudinal and Torsional Shafting Vibrations

(Program used by Maritime Administration.)

Number and/or Name

**C-9-002

Category(s)

**Hull Shafting Calculations

Descriptive Program Title

**Shaft Vibrational Analysis using Holzer Method

Source Activity

**Office of Ship Construction
**Maritime Administration
**Washington, D.C.

Engineer(s) Name-Code-Phone

**Richard Siebert, 721.21, 254-7048

Programmer Name-Code-Phone

**NAVSEC

Program Status

**Production

Classification (Security)

**Restricted - NAVSEC Program

Programming Language

**FORTRAN IV

Computer Type Used

**Control Data 6600

Special Hardware

**None

Special Hardware/Operation

**None

Program Size-Source Deck Cards

**258

Program Size-Object Core Words

**CM50000 Octal Words

Average Running Time (Min)

**2.57

Program Availability

**September 1970

Documentation Status

**Informal - Complete (15 pages)

Program Abstract

This program calculates torsional and longitudinal critical vibration frequencies using the Holzer Method. It was originally developed by NAVSEC for the IBM-7090 and subsequently converted to the CDC-6600 by the Maritime Administration. Double precision requirements were eliminated. Input requires hand calculation of all masses, inertias, and stiffness factors for each component in the turbine-gear-shaft-propeller system. Damping factors are not included in the calculation. Output consists of critical frequencies in CPS and RPM for various numbers of blades.

C-2. Longitudinal Shafting Vibrations

(Program used by J. J. McMullen Associates, Inc.)

Number and/or Name

**F-8-008

Category(s)

**Machinery Shafting and Bearing Calculations

Source Activity

**John J. McMullen Associates, Inc.
**One World Trade Center-Suite 3047
**New York, New York 10048

Engineer(s) Name-Code-Phone

**Engineering Division

Program Status

**Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**IBM 360/40 and IBM 1130

Documentation Status

**Informal - User's Guide

Program Abstract

Lumped mass system, using "level" effect. Text description found in NSRDC Report 3358, September 1970. Computes frequencies up to four modes.

C-3. Longitudinal and Torsional Shafting Vibrations
(Program used by Newport News Shipbuilding and Drydock Company.)

Number and/or Name

**F-0-016 9.5.0251 FORCE VIB

Category(s)

**Machinery Shafting and Bearing Calculations

Descriptive Program Title

**Longitudinal and Torsional Vibration in Propulsion
Shafting Systems

Source Activity

**Newport News Shipbuilding and Drydock Company
**Technical Systems Division
**4101 Washington Avenue, Newport News, Virginia 23607
(804) 247-7500

Engineer(s) Name-Code-Phone

**A. S. Pototzky

Programmer Name-Code-Phone

**F. E. Siegel

Program Status

**Active Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**Honeywell 6080

Program Availability

**Not available for general distribution

Documentation Status

**Incomplete

Program Abstract

FORCE VIB is a computer program to calculate the steady-state longitudinal or torsional vibratory response of branched shafting systems, such as propulsion systems. The system may have a maximum of 35 elements consisting of masses, dampers, and springs, all with only one degree of freedom. The masses and springs may be lumped or distributed, and the dampers may be viscous or solid. The program uses the mechanical impedance method to calculate displacements, forces, and phase angles, which may all be frequently dependent. The program also allows the varying of values to conduct parametric studies.

C-4. Longitudinal Vibration of Shafting, II

(Program developed and used by Littleton Research and Engineering Corp.)

Output Information

1. A plot of the blade-order harmonic force at the thrust bearing as a function of rpm.

2. A plot of the amplitudes of axial motion at the propeller and at the thrust bearing as a function of rpm.

3. Tabular data for above.

Input Information

1. Shafting arrangement, diameters, and lengths.

2. Propeller mass, diameter, number of blades, pitch, and developed area ratio (or mean width ratio).

3. Harmonic thrust (can be determined by program described in Appendix A-2).

4. The stiffness of the thrust bearing and its foundation.

5. Reduction gear weight.

Basis for Calculation

The propeller is represented by its mass plus entrained water and damping, estimated by Lewis and Auslaender's recommendations. The shaft is represented by a distributed mass and elasticity and is assumed to have a hysteretic damping (nominally 4%). The thrust bearing is represented as a concentrated mass elastically connected to a rigid hull.

This is a proprietary program not developed for general distribution.

APPENDIX D

COMPUTER PROGRAMS FOR COMPUTING THE LATERAL RESPONSE OF THE PROPULSION SYSTEM

D-1. Transverse Response of a Beam

(Program used by Newport News Shipbuilding and Drydock Company.)

Number and/or Name

**A-12-00 9.5.0301 Beam Vibration

Category(s)

**Conceptual Design
**Ship Vibrations

Descriptive Program Title

**Vibration Analysis of Beams

Source Activity

**Newport News Shipbuilding and Drydock Company
**Production Computer Systems Division
**4101 Washington Avenue, Newport News, Virginia 23607
(804) 247-7500

Program Status

**Production Use

Classification (Security)

**Unclassified

Programming Language

**FORTRAN V

Computer Type Used

**Honeywell 6080

Special Hardware

**None

Program Availability

**Not available for general distribution

Program Abstract

Computes the steady-state transverse vibratory response of a beam with any number of intermediate flexible supports, with generalized end conditions, section properties, and loading.

D-2. Transverse Vibration of Shafting and Propeller

(Program developed and used by Littleton Research and Engineering Corp.)

Output Information

1. Plots of bearing forces, in two normal directions, as a function of frequency.

2. Plots of shaft motions, in two normal directions, as a function of frequency, at the propeller and at other critical locations.

3. Plots of the shaft deflection curves at each natural frequency within the operating speed.

4. Plots of the steady plus harmonic bending moments in the shaft of the aftermost bearing.

5. Computer tables for above.

Input Information

1. Shafting arrangement, diameters, and lengths.

2. Propeller weight and moment of inertia about its rotation axis, diameter, number of blades, pitch, and developed area ratio.

3. Stiffness or flexibility matrices for each bearing about axes perpendicular to the axis of rotation (force and rotation).

4. Horizontal and vertical harmonic forces and moments, and the steady thrust (can be determined by the program described in Appendix A-2).

Basis for Calculations

The propeller is represented by its mass, its entrained water, its moment of inertia about the rotational axis, its moment of inertia about an axis perpendicular to the rotational axis, the moment of inertia of its entrained water, and the hydrodynamic damping in its several modes of motion. The shaft is represented as a series of uniform beams having distributed mass and bending stiffness and hysteretic damping. The bearings are represented by their stiffness in translation in two directions mutually perpendicular to the shaft axis and by their stiffness in rotation about the same two axes. The bearings are assumed to bend with the shaft; however, where there is flexibility between shaft and bearing, e.g., rubber staves, this flexibility, lateral and angular, is incorporated in the strut matrix. It

is generally acceptable to terminate the shaft at the after inboard line-shaft bearing.

The program computes the vibration in terms of coupled properties in the horizontal and vertical directions. It includes the influence of the steady thrust (small effect), but not that of the steady torque (very small).

This is a proprietary program not developed for general distribution.

APPENDIX E

COMPUTER PROGRAMS FOR COMPUTING THE RESPONSE OF THE ENTIRE HULL GIRDER STRUCTURE

E-1. General Bending Response Program

(Program developed and used by Naval Ship Research and Development Center.)

Name

**GBRP

Category(s)

**Lateral, Longitudinal, and Torsional Beam Vibrations
**Bending Coupled with Torsional Beam Vibrations
**Whirling Vibrations of Propeller Shafts

Descriptive Program Title

**General Bending Response Program

Source Activity

**Naval Ship Research and Development Center
**Bethesda, Maryland 20084

Engineer(s)

**Michael E. Golden
**Francis M. Henderson

Program Status

**Active Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**CDC 6000 Series

Output Plotting

**SC 4020 Plots

Special Software/Operation

**Overlays for Program Subroutines
**Open Core to Optimize Storage

Program Availability

**Availability for general distribution through David Taylor Naval Ship Research and Development Center

Documentation

**Complete

Program Abstract

The General Bending Response Program (GBRP) consists of the union of three programs: General Bending Response Code 1 (GBRC1) for lateral, longitudinal, and torsional vibrations; GBRC2 for vibrations involving bending coupled with torsion; and GBRC3 for whirling vibrations of propeller shafts. The latter two codes resulted from an extended application of the mathematical model used in the first code. The program formulates the finite-difference equations which approximate the boundary-value problem representing the steady-state motion of a vibrating nonuniform mass-spring system such as a ship hull or shafting in bending. The program calculates natural frequencies and mode shapes and the response to specified harmonic driving forces and moments. The program can represent a ship hull connected elastically to other systems such as the propulsion system and to sprung masses. Longitudinal or torsional vibrations problems can also be solved by dividing each beam into sections connected by springs, thus reducing the model to a mass-spring system. See Reference 73.

E-2. Ship-Hull Vibratory Response

(Program used by USS Engineers and Consultants, Inc.)

Number and/or Name

**A-12-006 SHRVS

Category(s)

**Conceptual Design
**Ship Vibrations

Descriptive Program Title

**Simulated Ship-Hull Vibration

Source Activity

**USS Engineers and Consultants, Inc.
**600 Grant Street
**Pittsburgh, Pennsylvania 15230

Engineer(s) Name-Code-Phone

**F. Ronald Griffith
(412) 433-6517

Program Status

**Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**CDC 6500

Special Hardware

**None

Special Software/Operation

**None

Program Size-Source Deck Cards

**4,000 cards

Program Size-Object Core Words

**160K Octal Words

Average Running Time (Min)

**3 minutes

Program Availability

**Time Sharing Service
Call F. R. Griffith

Cost

**Negotiable

Documentation Status

**Informal Complete

Program Abstract

The purpose of SHRVS is the accurate prediction of the vibration response of a ship hull to either steady-state or transient loads applied in the vertical centerline plane of the hull. Factors considered include cargo distribution, bulkhead location, machinery space location, as well as the flexural and shear stiffness of the main-hull girder and of the double-bottom structure.

APPENDIX E-3

COMPUTER PROGRAMS AVAILABLE FOR COMPLETE HULL ANALYSIS

	Lloyd's Register of Shipping	Netherlands Ship Model Basin	Institut für Schiffstechnik Technische Universität Berlin	Electric Boat Division, General Dynamics Corp.	Computas, Subsidiary of Det norske Veritas
NAME OF PROGRAM	Hull Vibration LR57P	DASH	FREIS, ERZS	GENSAM	SESAM-69
PERSON TO CONTACT	Geoffrey H. Sole	Dr. S. Hylarides	Dr. E. Metzmeier	Dr. Henna Allik	B. Aamodt
TYPE OF ANALYSES	Eigenvalues, Eigenvectors	Eigenvalues, Eigen- vectors, Steady-State, and Transient Response	Eigenvalues, Eigen- vectors, Steady-State, and Transient Response	Eigenvalues, Eigen- vectors, Steady-State, Transient, and Random Response	Forced and Free Vibration Analyses of Steady-State or Transient Vibrations
DATE OF INITIAL COMPLETION/LATEST REVISION	U/1976	1970/1975	1974/U	1967/1976	1968/1975
AVAILABILITY/ PRICE	Service from Lloyd's Register	Service from NSMB	U	Proprietary - Avail- ble on a case basis	Service use available for rent or sale
LANGUAGE	FORTRAN IV	ALGOL	FORTRAN	FORTRAN V	USASI FORTRAN
COMPUTER SYSTEM	IBM 370/158	Irrelevant	CDC 6500/ Scope 3.4	UNIVAC 1106, 1108, 1110 EXEC 8	UNIVAC 1108, Other Computers
BASIS OF PROGRAM	U	Finite Element	Finite Difference	Finite Element	Finite Element
TYPE OF STRUCTURES	Represented by a Free-Free Beam	Complete or Partial Ship	Ship Hull	General Structures	General Structures
MANNER OF REPRESENTATION	Beams	Beams, Plates	Beams	Beams, Plates, Shells, 2- and 3-D Continua	Beams, Plates, Shells, 2-D and 3-D Continua
CONCENTRATED OR DISTRIBUTED MASS	Concentrated	Concentrated	Concentrated	Both	Concentrated
SUBSTRUCTURING	No	U	U	Yes	Yes
COMPRESSION OF DYNAMIC MATRIX	No	No	U	Yes	Yes
TREATMENT OF WATER INERTIA	Direct Input or Internal Computation	Direct Input	According to Lewis and Kumai	U	Direct Calculation
TREATMENT OF DAMPING	No Damping	Viscous	Viscous and Hysteretic	User Supplied Damping Arrays Except Viscous	Modal Damping
PROGRAM CAPACITY	121 Nodes 240 DOF	Unlimited	41 DOF Long. 82 DOF Vert. 164 DOF Coup. Hori.-Torsion	Unlimited	Unlimited

U = undefined

APPENDIX F

LONGITUDINAL, TANGENTIAL, AND AXIAL WAKES FOR THE CONTAINERIZED AND UNITIZED CARGO SHIP ANALYZED IN CHAPTER V

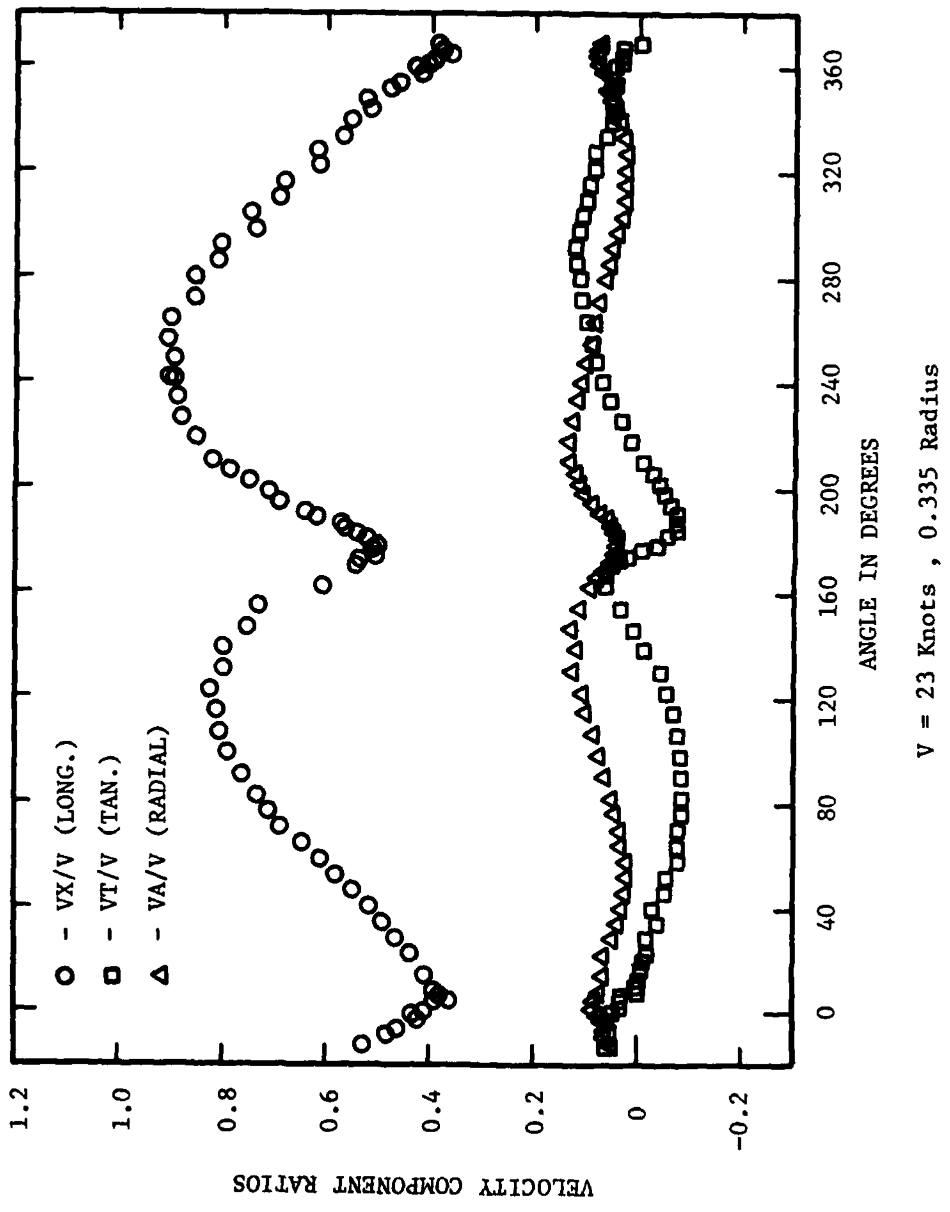


FIGURE F1. LONGITUDINAL, TANGENTIAL, AND AXIAL WAKES AT 0.335 RADIUS

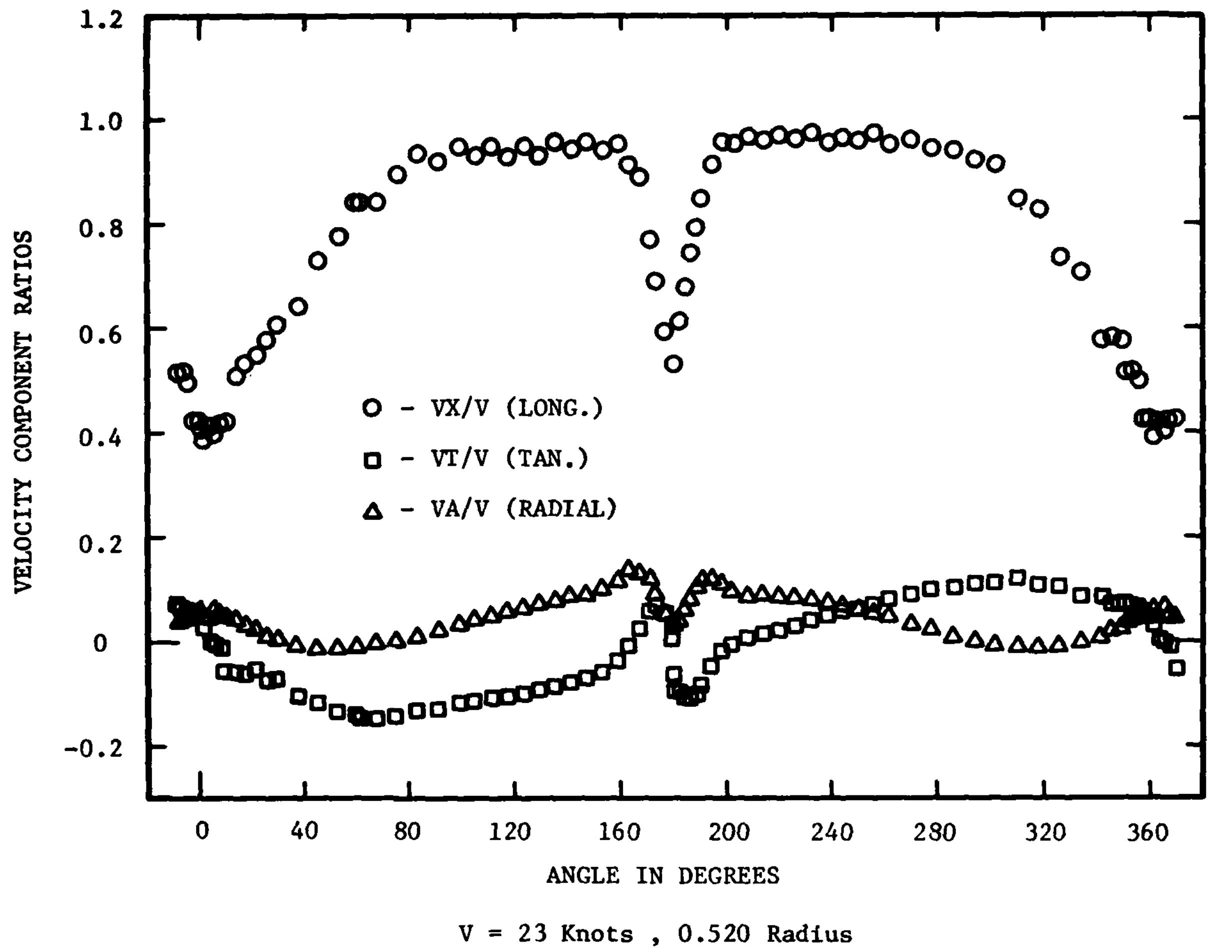


FIGURE F2. LONGITUDINAL, TANGENTIAL, AND AXIAL WAKES AT 0.520 RADIUS

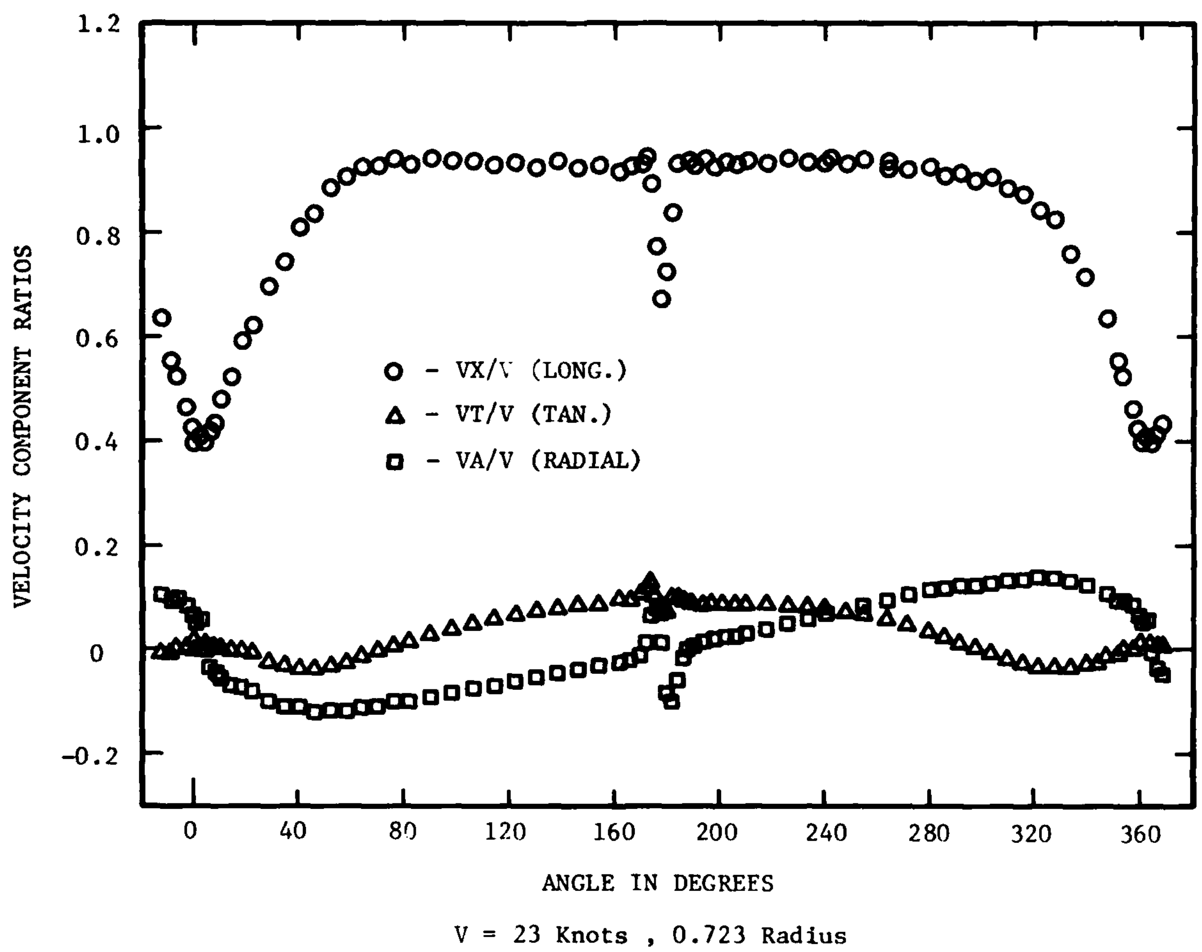


FIGURE F3. LONGITUDINAL, TANGENTIAL, AND AXIAL WAKES AT 0.723 RADIUS

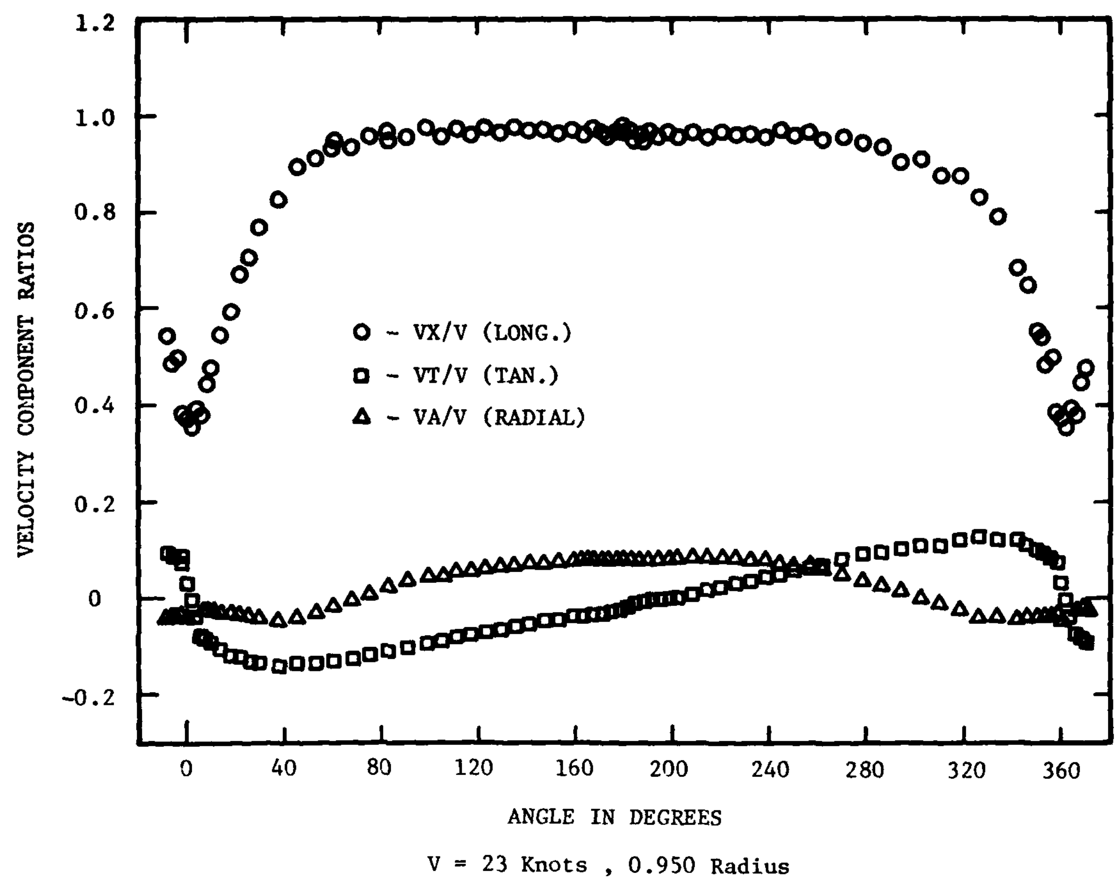


FIGURE F4. LONGITUDINAL, TANGENTIAL, AND AXIAL WAKES AT 0.950 RADIUS

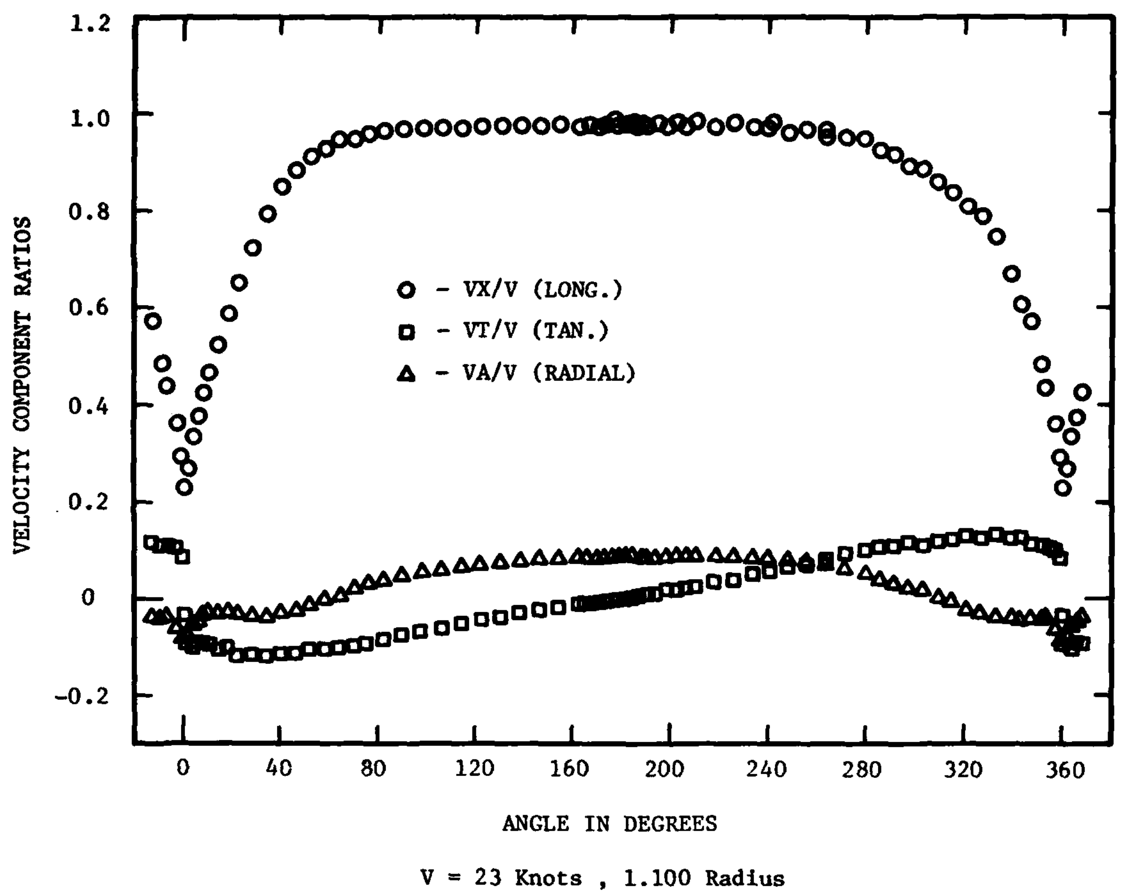


FIGURE F5. LONGITUDINAL, TANGENTIAL, AND AXIAL WAKES AT 1.100 RADIUS

APPENDIX G

SUMMARY OF VIBRATION STUDIES CONDUCTED BY LITTLETON RESEARCH AND ENGINEERING CORP. FOR AN RO/RO TRAILER SHIP DESIGNED BY SUN SHIPBUILDING AND DRYDOCK COMPANY

Rept. No.	Title	Summary
I	Vibratory Propeller Forces, Hull Pressures and Forces, and Propeller Stresses of a Trailer Ship	Report of computation of propeller forces transmitted to shafts, hull pressures, and hull forces for center propellers and wing propellers having 4, 5, and 6 blades.
II	Longitudinal and Transverse Vibration of Shafts, Sun Trailership, Hull 665	Natural frequencies of ℄ and wing shafts in longitudinal vibration as a function of combined thrust bearing foundation stiffness. Resonant frequencies of ℄ and wing shafts in vertical and transverse vibration. Hull assumed rigid, strut stiffness in translation and rotation included, bearing fluid film assumed rigid.
III	Transverse Vibration of Shafts, Raked and Unraked Struts, Sun Trailership Struts	Studied the effects of changes in shaft diameter on lateral frequency. Considered the influence of strut rake on vertical and transverse resonant frequencies
IV	Calculated Static Stiffnesses of Thrust Bearing Foundations and Longitudinal Vibration of Shafts, Sun Trailership, Hull 665	Determined the static stiffness of the foundation of the ℄ and wing thrust bearing foundations using a finite element analysis of the engine room structure. Using these stiffnesses and a stiffness of the thrust bearing, the natural frequencies of ℄ and wing shafts were computed.
V	Frequencies and Mode Shapes of Shafts and the Machinery Space, Sun Trailership, Hull 665	Using a dynamic finite-element representation of the shafting and adjacent hull structure, the shaft vibration frequencies of the ℄ shaft system in vertical motion and the wing shaft in vertical and transverse motion were determined. These are significantly lower than

Rept. No.	Title	Summary
V		those determined under the rigid-hull assumption. Longitudinal vibration frequencies of the ℄ and wing shafts are determined. It is found that the fundamental frequencies are not much changed from those obtained using static stiffness, but that new frequencies of importance are introduced.
VI	Calculated Propeller Induced Vibration Levels, Sun Trailership, Hull 665	The vibration levels throughout the ship are predicted. Levels are reasonable by ISO standards in the machinery operating stations and in the crew quarters low in the hull. However, they are excessive in the house. Frequencies of peak amplitudes generally coincide with shaft resonance frequencies.
VII	Effect of Center Thrust Bulkhead Stiffening on Longitudinal Shaft Frequency, Sun Trailership, Hull 665	Several thicknesses of the longitudinal bulkheads supporting the ℄ thrust bearing are studied to determine the influence on longitudinal natural frequency.
VIII	Effect on Relocating the Center Thrust Bearing on the Longitudinal Shaft Resonance, Sun Trailership, Hull 665	In an effort to increase the natural frequency of longitudinal vibration of the ℄ propeller shaft system, the effect of moving the thrust bearing aft is studied. This modification is found to be more effective than increasing the scantlings of the thrust foundation structure.
IX	The Effect of Design Changes Upon the Resonance Frequencies of Propulsion Shafts, Sun Trailership, Hull 665	The design changes are made in an attempt to bring the shaft resonance above the blade frequencies of a 5-bladed ℄ propeller and 4-bladed wing propellers. Structural changes such as increasing thickness of a longitudinal bulkhead, adding a transverse bulkhead, increases in shaft diameter, modification of struts, increases in thickness of shell plating and plating of a flat, plus a change in location of wing propellers, are considered.

Rept. No.	Title	Summary
X	Guide for Response Output Interpretation, Sun Trailership, Hull 665	Users' Guide
XI	Effect of Stern Bearing Location and Strut Rake on Center Shaft Vertical and Lateral Vibration, Sun Trailership, Hull 665	Using the simple shafting model, several alternative processes for changing the transverse frequency without changing the vertical frequency are explored.
XII	Center Shaft and Strut Vibration Study, Sun Trailership, Hull 665	By reducing propeller overhang, moving the struts aft on the barrel, stiffening the hull, and increasing the strut thickness and chord with tapering, it is possible to locate the vertical and transverse frequencies of the centerline shaft at desirable values.
XIII	Wing Shaft and Strut Vibration Study, Sun Trailership, Hull 665	It is shown that a combination of scantlings and rake for the wing propeller struts can result in acceptable resonant frequencies. The addition of a transverse bulkhead to avoid internal hull resonance is recommended.
XIV	Vibration Evaluation of Built-up Struts, Sun Trailership, Hull 665	Previous studies have been based on solid cast steel struts. Because of procurement difficulties, fabricated struts are of interest. Suitable built-up struts are determined.
XV	Vibration Evaluation of Alternate Strut Designs, Sun Trailership, Hull 665	The design of the struts was shown in previous studies to have a strong influence on the shafting resonant frequencies. This study investigated the effects upon natural frequency of changing strut rake and different rakes in each of the strut arms.
XVI	Static Deflections and Stresses, Sun Trailership, Hull 665	Use is made of the finite-element model, developed for dynamic analysis, to compute the hull deflections due to shaft thrust and torque. These deflections are evaluated against limits set by the reduction gear manufacturer.

Rept. No.	Title	Summary
XVII	Calculated Propeller Induced Vibration Levels, Sun Trailership, Hull 665	This is the final update of the whole hull vibration level prediction. As a consequence of the attention given in the shaft resonances, the vibration level at design power is at a minimum with peaks below and above. Although the vibration at the lower peak would not be acceptable under continuous operation, this frequency is at a speed which would be seldom used.

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The Ship Design, Response, and Load Criteria Advisory Group prepared the project prospectus, evaluated the proposals for this project, provided the liaison technical guidance, and reviewed the project reports with the investigator:

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